Heating Piping & Air Conditioning

with which is merged the Aerologist

Design, Installation, Operation, and Maintenance: Industrial, Commercial, Institutional, and Public Buildings

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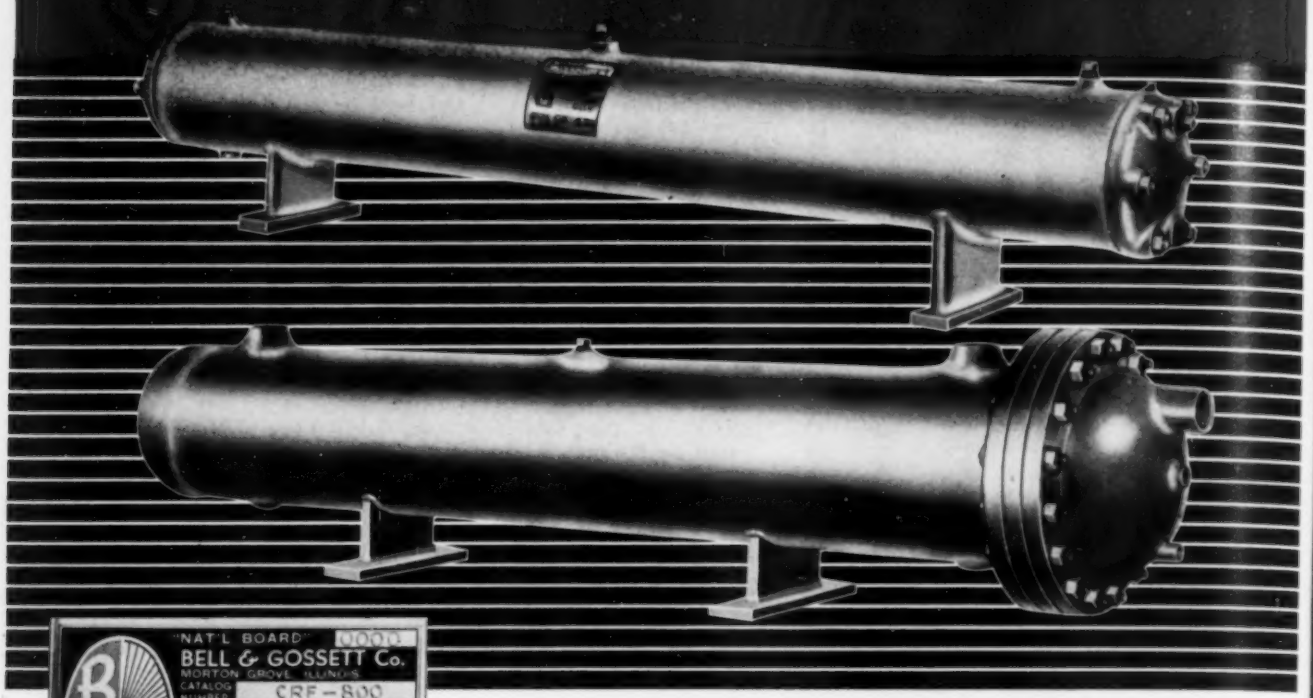

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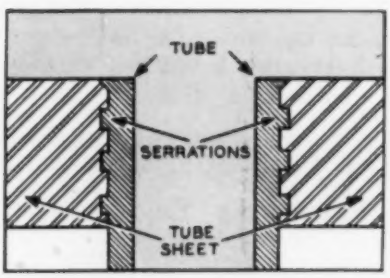

The diagram above shows how the tube wall is forced into serrations, or grooves, in the tube sheet when the tube is rolled in. A leak-proof junction between tube and sheet is thus effected.

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THE EDITOR'S PAGES

JANUARY BUILDING CONTRACTS BREAK

Contracts awarded for building and heavy engineering projects in the 37 states east of the Rocky Mountains in January showed a sharp break from the volume reported in December and in January of last year, it has been reported by the F. W. Dodge Corp.

A statistical summary of investment commitments reported in field reports showed a total contract volume for January of $482,984,000, 31 percent less than that reported in December and 22 percent lower than the volume reported in January of last year.

The overall decline in building and construction contracts was least severe in New England, the middle Atlantic, and southeastern states, and the region comprised of northern Illinois, Indiana, Iowa, Wisconsin, and northern Michigan.

The decline was especially pronounced in residential and heavy engineering awards. Increases in awards for hospitals, institutional buildings, and churches lessened the severity of the drop in nonresidential buildings as a group. The volume of nonresidential awards was $221,883,000, 17 percent less than in December and 8 percent less than in January of last year.

PREDICTS 5 TO 10 PERCENT CONSTRUCTION DECREASE

A 5 to 10 percent decrease in the volume of construction from last year's all-time high, resulting in stabilization or even reduction of building costs, was forecast for 1949 by Melvin H. Baker, president of the National Gypsum Co., in an address at the 96th annual meeting of the American Society of Civil Engineers.

"The output of most materials has been increasing steadily since the close of the war," Mr. Baker said, estimating the 1948 production at 5 percent higher than that of 1947, in which was "recorded the highest output of any year since 1915."

He attributed the 117 percent increase in the composite price of all construction materials in June 1948 over the composite price in 1939 to the "record breaking demand and the impact of postwar inflationary forces."

TESTS GLYCOL FOR AIR STERILIZING

Sterilization of air in offices with glycol vapor will be tested in the Chicago Tribune's telephone adtaking area, Dr. Theodore Van Dellen, head of the Tribune medical department, disclosed recently.

Health records maintained by the Tribune will enable the medical department to compare the time lost through sickness during the test period with records for the same period last year. Sterilizing of air throughout the entire new eight-story annex to Tribune tower may be attempted if an appreciable improvement results.

Liquid glycol, vaporized as it spirals around a heated pipe and distributed throughout the room by a fan, is expected to kill many harmful bacteria. Some medical authorities suggest that the glycol may protect the workers for a short while even after they have left the office.

Equipment to be used in the test is a little bigger than a standard typewriter case and is operated by electricity from an ordinary wall socket. Dr. Van Dellen hopes to attach such a device to the air conditioning system in order to sterilize all the air in the Tribune tower annex as it is circulated, if the test indicates that the glycol vapor is effective.

* * *

OVERHEARD AT A TECHNICAL MEETING: "The mind can absorb only what the seat can endure."

SUGGESTION FOR A CORPORATE NAME: The representative of a control manufacturer said at a recent meeting of architects that he had suggested to the management of his company that they change their name to "Approved Equal." He felt that this would simplify greatly his job of getting his equipment specified.

FRONT COVER PHOTO

The picture on the front cover is a view of the plenum chamber over the main floor of the manufacturing building at the new electronics plant of the Western Electric Co., Allentown, Pa. The ducts on the floor of the chamber lead to the air diffusers, which are directly under the circular baskets. The baskets are set over the return air openings of the diffusers to guard against anything being accidentally dropped through to the floor below.

There are over 500 specially designed and adjusted air diffusers. Those for the main floor of the manufacturing building are designed to return a controlled volume of air to the overhead plenum for reconditioning and recirculation or exhaust. Particularly unique in this respect are the diffusers serving the central area of this main floor because, in order to satisfy the prescribed balance, the air supplied to the peripheral areas must flow toward the center area. This required the 208 diffusers in the central area to extract from it approximately 30 percent more air than they supply to it.

The outer cone of these diffusers has an outside diameter of 45 in., or almost 4 ft. The supply neck is 27 in. OD and the return neck 19⅞ in. OD, with supply and return areas of 1.82 sq ft and 2.15 sq ft, respectively. It is believed that this is the first time that a combination supply and return air diffuser has been designed with greater return than supply air capacity. (Photo courtesy W. B. Connor Engineering Corp.)

THE EDITOR'S PAGES

WHY BOILER EFFICIENCY IS SO IMPORTANT

The reason boiler efficiency is important was stated simply and effectively by Fred Hainer, of the Cleaver-Brooks Co., in a talk on packaged steam generators at the February meeting of the Illinois chapter of the American Society of Heating and Ventilating Engineers. "You buy a boiler once," Mr. Hainer said, "but you put through it during its life many, many times its cost in fuel."

WHAT MAKES A GOOD INSTALLATION

"Good design, a good contractor, and good instructions to the operating men make a good heating or air conditioning installation," according to John Dolio, of Shaw, Metz and Dolio, architects, Chicago, in his introductory remarks at a session on heating and air conditioning at the first regional meeting of the north central district of the American Institute of Architects. His firm's designers include as part of their work the writing of operating instructions to turn over to the owner.

There is an increased demand from the public and from owners for "more perfectly functioning" heating, ventilating, and air conditioning systems, he said. "If the temperature varies more than a couple of degrees, someone gets called about it these days," he pointed out. In smaller buildings particularly, the more automatic is the system, the less trouble is encountered with it, he remarked.

SEE GAIN IN STOKER SALES

Most of the country's manufacturers of underfeed stoker equipment—together with representatives of the industry's major supply firms and guests representing allied industries and the press—met in Chicago January 27, for the winter membership conference of the Stoker Manufacturers Association.

While the meeting considered and acted upon a number of routine matters of immediate interest to the stoker trade and industry, an unofficial poll taken of the members indicated that many manufacturers anticipate increases ranging from moderate to substantial gains in their stoker sales volume in 1949 as compared with 1948, when approximately 90,000 stokers were sold by manufacturers to distributor and retail outlets. The general average increase indicated was about 33 percent on a unit basis with a possibility of the dollar volume exceeding this percentage increase slightly—depending on the need and demand for stoker equipment used in commercial and industrial plants and establishments.

FORM INTER-SOCIETY CORROSION COMMITTEE

Fifteen delegates and three observers from technical organizations concerned with corrosion met recently at the headquarters of the American Society for Testing Materials, Philadelphia, to formulate the organization of the Inter-Society Corrosion Committee of the National Association of Corrosion Engineers.

The new committee's objectives were outlined in part during the meeting as follows:

1) To promote cooperation among various technical societies working in the field of corrosion.

2) To prepare and publish periodically a directory of corrosion workers in the United States.

3) To provide liaison among technical committees of the NACE and other technical societies.

4) To act in advisory capacity to technical societies interested in corrosion in an effort to promote progress of all and avoid duplication of effort.

5) To recommend in which society a proposed new activity might be carried out best without, however, attempting to define or limit the field of activity of any society.

6) To stimulate the publication of reliable data on corrosion and to provide publicity for conferences in this field.

The inter-society committee will carry on some activities of the former American Coordinating Committee on Corrosion and the NACE committee on relations with other technical societies, both of which it has succeeded.

A. B. Campbell, 905 Southern Standard Bldg., Houston 2, is executive secretary of the NACE.

15 Years Ago

The lead-off article in the March 1934 HPAC, under the title Air Conditioning Fast Becoming Standard Equipment for Railway Passenger Cars, described the heating and air conditioning of the Union Pacific's new M-10,000, pioneer streamliner completed at the Pullman Car & Mfg. works the month before. The train, "painted a golden brown and canary yellow for visibility" comprised three cars, articulation permitting it to be carried on four trucks instead of six. The main power plant consisted of a 600 hp. distillate burning, internal combustion engine.

* * * * *

Paul Bancel discussed steam jet units for water vapor refrigeration in his article in the March 1934 HPAC. This was the first of a two-part article, the second of which was devoted to centrifugal water vapor machines.

* * * * *

Sabin Crocker's article on the new code for pressure piping in the March 1934 HPAC was devoted to section 3 on oil piping systems for the production, transmission, and refining of petroleum.

* * * * *

In a short article in the March 1934 HPAC, Mrs. Snyder, operator of 13 candy shops in Chicago and its suburbs, spoke of the excellent results achieved with air conditioning installations in two of them. predicted that air conditioning "will soon be common-place in up-to-date stores."

* * * * *

Chapter I of the NRA code of fair competition for the construction industry (remember the "Blue Eagle"?) became effective on the 2nd of the month, it was noted in the March 1934 HPAC.

"OPEN for DISCUSSION"

DO YOU HAVE something to say about any of the articles published this month? An additional comment, or perhaps a question on some point? We follow here the custom of engineering society meetings in allowing a period for discussion following presentation of a paper. You are invited to contribute your views. Address them to the Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2

WHY DOES OIL BURNER FLAME DIE DOWN

IT SEEMS to me that Kalman Steiner, in his answer to the question "Why Does Oil Burner Flame Die Down?", on page 104 of the December issue, has completely missed the No. 1 cause for the effect produced—said cause not requiring a knowledge of tank location, foot valve, etc. for the poor service-man or trouble-shooter to dig out at a cost of $3.50 to $5.00 per hr before being able to diagnose the trouble and fix it.

Being called in for service, job unknown, to straighten out a difficulty such as stated in the question, and not knowing the type of burner, type of system, type of pump, type of control, method, size and condition of piping, location of tank, when last cleaned, when last serviced, etc., no one can immediately lay a finger on the cause and accomplish the remedy. However, if a client asked me the question, I would immediately assume that the job had been working O.K. previously and would consequently answer (or instruct the service-man) as follows:

"Burning No. 6 oil requires that the oil be preheated (the most important auxiliary on heavy oil installations is the fuel oil preheater). These preheaters use as a heating medium a steam or hot water heater or electric heating elements thermostatically controlled. The heating tubes or elements gradually get coated with the heavy tars out of the oil. This coating acts as an insulator in that it retards the rate of flow of heat through the surface into the oil. To maintain the flame the burner requires a certain number of gallons of oil per hour which is a set rate of flow, and which in turn requires a set rate of heat supply to that flowing oil.

"If the rate of heat supply is cut down by the coating of insulation on the tubes or elements, the oil rate is not only cut down but the increased viscosity decreases the flow through the nozzle, thereby cutting the firing rate further. The fact that the fire burned O.K. for 10 to 15 minutes after lighting would tend to show that sufficient heat was available, but that after the residual heat was used up the rate of heat flow was not sufficient to maintain the rate of oil flow. Check the heater surface, elements and thermostats—if you find them O.K., call me."—W. H. HERMES, III, consulting mechanical engineer, San Diego.

RADIANT HEATING DESIGN CONTROVERSIAL

QUITE APPARENT it is that there is little agreement among individuals as to the proper method of design of radiant heating systems. H. W. Schiewe is quite correct when he says (on page 67 of the February issue) that the calculation is complex because of the variables that enter the problem.

If Mr. Schiewe accepts Dr. Giesecke's calculations and gets results that are satisfactory, that seems to be the desired result. However, D. L. Mills says (on page 104 of the February issue) that investigation of laboratory and field panels indicates nothing that would lead to a verification of the theory that 3.5 Btu per sq ft of pipe surface per F per hr is a correct factor.

The purpose of the article by J. K. Fairbairn and myself in the January issue was not the introduction of a *U* factor nor the discussion of one. We presented a chart which would simplify the calculation after the individual designer had selected his own *U*. Our selection of a *U* was for the purpose of setting up an illustrative problem.

We are now in the process of developing a nomograph chart based on our theory of the *U* coefficient. If heat flow information published by F. W. Hutchinson, Dr. Giesecke, or others is being used, our nomograph should prove very useful and accurate (for the *U* used) for final panel design.

Existing radiant systems when tested do not balance out to the predicted designed drop. They work and heat to the owners' required comfort but the drop is much greater than calculated.

The *U* which we used in the example in our article, and panels designed with our chart, have been checked very successfully in the field by thermocouples for surface temperatures and temperature drops.—R. P. GOEMANN, John D. Dillon, consulting engineer, New York City.

EFFECT OF ENTRANCE ON FLOW OF FLUIDS

MY REPLY to Marcello Giomi's comments (on page 84 of the February issue) regarding my article on piping fundamentals, which was published in the December issue, is as follows:

The flow of water to the outlet of a tank follows a perpendicular line; that is, if the tank is being constantly refilled. If the tank is being emptied, there will be a horizontal flow of water to replace the water leaving the tank, but the flow to the outlet will be in a vertical line.

To make the foregoing statement more clear, let us analyze the "latent force" of the water prior to the start of flow. The line of pressure will be vertical due to gravitation. When the flow begins, naturally the force directly in a vertical line with the opening will be exerted to induce the water through the opening. Lines of force follow the lines of least resistance; therefore, the line of force will be vertical and not horizontal or even oblique.

In the design of heat exchange equipment we have found, and usually the hard way, that the flow of fluid toward an outlet is in a vertical line and practically no circulation is induced except in a narrow circular area extending vertically upward from the outlet connection. —HERMAN VETTER, consulting engineer, Los Angeles.

"OPEN for DISCUSSION"

FIRE HAZARDS IN WINDOWLESS BUILDINGS

REFERRING TO my article on Eliminating Fire Hazards in Windowless Buildings, page 86 of the February HPAC, and George W. Booth's comment given in the editor's note on the same page, I believe Mr. Booth's suggestions certainly contribute to an understanding of the problem and are based on the experience of many years in fire prevention engineering. However, the amendment to the National Board of Fire Underwriters standards for the installation of air conditioning, warm air heating, air cooling, and ventilating systems as recommended by the National Fire Protection Association at its last annual meeting, makes no mention of automatic sprinkler protection in the new paragraph 103 on mechanical smoke removal. This paragraph reads:

"103. It is recommended that in buildings, especially of the windowless type, where the panic hazard is pronounced, means be provided for mechanically removing smoke. Such smoke removal systems may incorporate such portions of air conditioning, heating, ventilating or exhaust systems as may be adapted to the purpose. It is important that the system be arranged to keep smoke away from exitways.

"In the absence of more definite data, it is recommended that the smoke exhaust equipment have a capacity approximating a 10 minute air change."

The National Building Code, however, does set up requirements for automatic sprinklers in certain types of windowless buildings.

There is still some difference of opinion among fire underwriters as to the desirability of venting fires. In fact, I originally voted in committee against the proposal, but the majority of the committee (some of whom had studied the matter in considerable detail) favored the proposal.

In my article I quoted from the bulletin of the Associated Factory Mutual Fire Insurance Cos., as follows:

"This may increase the rate of burning somewhat by introducing currents of fresh air, but the effect would not be much different than if windows and doors were opened to vent smoke. Venting is necessary if fire is to be fought effectively."

This seemed to be the attitude of the majority of the members of the committee on air conditioning.—HOWARD C. MURPHY, consulting engineer, Carmel, Calif.

USE CARE WHEN YOU USE CARBON TET

IT SEEMS appropriate that the two following paragraphs should be mentioned in connection with the article on suggestions for cleaning ducts, which was published last year in HPAC. The information is copied from the reference on Carbon Tetrachloride-Gaseous Products of Decomposition of the card data service, Underwriters' Laboratories.

"In the vapor form, carbon tetrachloride is an anesthetic having an inoffensive odor. It is classified as to toxicity in Group 3, being lethal to guinea pigs in concentrations of the order of 2 to 2½ percent by volume for continuous durations of exposure of the order of 1 hr in closed rooms. (No ventilation.)

"Dry cleaning solvent. When carbon tetrachloride is used for dry cleaning in small rooms or confined spaces, there is the possibility of more or less continuous or recurrent exposure of the operator to the vapor in concentrations injurious to health unless adequate ventilation is provided, or else suitable equipment is employed to confine the vapors within the unit used."—MARVIN GOULD, assistant professor, Iowa State College, Ames.

ANCIENTS USED PREFERRED NUMBERS FOR PIPE SIZES

That "there's nothing new under the sun" has its element of truth is interestingly demonstrated in an article by Edmund A. Pratt published in the February issue of Standardization, news magazine of the American Standards Association, Inc. He shows that the ancient Romans apparently used "preferred numbers"—a fairly modern idea with a distinct factory background. Pipe fittings for connections to Roman aqueducts and reservoirs closely followed the geometrical progression of sizes now used in preferred number series.

Pipes were numbered according to their inside diameters in *quarter-digits,* the digit being 1/16 of a Roman foot. Pipe capacities were measured in *quinariae,* the quinaria being equal to the capacity of a pipe five quarter-digits in diameter. This was the base size, according to Mr. Pratt, and the other sizes were related to it in such intervals as to give a suitable range to meet practical needs.

By means of a plot of Roman pipe size ratios and preferred numbers, Mr. Pratt shows that the men responsible for the water supply of ancient Rome arrived some 1900 years ago at a "very modern conception of a suitable range of pipe sizes."

REPRINTS OF OPERATING EXPERIENCE ARTICLE READY

John H. Clarke's extensive series of articles entitled Design Engineer Reports on Operating Experience, publication of which was completed last year, have been reprinted by the author and copies are available from him for 75c each. These articles covered both engineering and other problems and how some of them were solved during the author's tour of duty on a tanker.

Many requests for reprints of these articles have been received, and have been referred to Mr. Clarke. If you want one or more reprints, just send your request and the necessary remittance to the Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2, and it will be forwarded on to Mr. Clarke.

Now available at no cost to subscribers is the volume index for Heating, Piping & Air Conditioning for the year 1948. If you have not already asked that this index be sent you, let us know and we'll mail one.

How to Design the Piping for Conveying Flashing Hot Water

THE AVERAGE engineer needs a lot of help in applying theory to the design of piping systems for conveying flashing mixtures of steam and water. This article has been written with the objective of giving the help required in solving everyday problems. The theory is reviewed in the appendix in order that the method may be understood. Curves are also presented from which precalculated data may be obtained. Without these curves it would be necessary to perform several graphical integrations, and to resort to cut and try methods to solve even the simplest of problems. Sample problems are given to demonstrate the calculation of pressure drop in four types of problems. In addition, a theory is developed for calculating surge pressures resulting from the high velocities which are usually encountered in condensate lines. Finally, suggestions are given for designing flash tanks.

Flashing Causes Big Volume Increase

Many of the difficulties encountered in the design of piping systems for conveying hot water are caused by the flashing which occurs in the pipe. As the fluid travels down the line, there is naturally a drop in the pressure. As a result of the drop in pressure, the water finds itself supersaturated; and, in order to maintain equilibrium, a portion of the water flashes into steam. Relatively only a small portion of the water flashes into steam; but there results from even a small amount of flashing a very considerable increase in volume. The small portion of the water which flashes may increase its volume over a thousandfold.

It follows that, as the volume increases, the velocity also increases—such that it often reaches very large values. A calculation which properly accounts for this considerable increase in velocity, as the water progresses down the pipe, is necessarily complicated. Because of the many difficulties encountered in designing even a relatively simple condensate line, it is quite apparent that the usual problem would formerly not justify the expenditure of the time required for a solution. The result has been that many designers have been forced to overdesign in order to get a safe answer. In the past, it has been cheaper to make the pipe plenty big than it would be to calculate an accurate solution.

Too Large a Pipe Is Objectionable

A careful study will show, however, that a pipe which is too big is apt to carry a higher velocity throughout a greater length than a line which is properly designed. The reason for this is in the critical flow effect which is almost always encountered.

Consider a system which has a trap or control valve orifice installed at the inlet end of the line. The high pressure drop in the line will serve to reduce the amount of flashing so that, as one moves even a short distance back from the outlet end, the velocity will be found to be greatly reduced.

Now observe what happens if we deliberately enlarge the diameter of the pipe. It simply means that more of the pressure drop has to be taken by the orifice. This means that much more of the water has flashed into steam at the inlet end of the pipe. The mass flow through the line will be increased a little, but the volume flow through the piping will be increased a great deal.

In the extreme case of a very large increase in the size of the piping, the pipe becomes in effect a long flash tank—and we would observe critical velocity only at the outlet of the orifice. In such a system, we would find very little pressure loss in the piping. For all practical purposes, however, we cannot get less than critical pressure at the outlet end of the piping. It can be shown that increasing the size of the pipe over that which is required is very likely to increase—not decrease—the velocity of the fluid in the piping. The result of increasing the size of the pipe has been higher first cost and shorter life. It is believed that many of the troubles encountered with condensate lines

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THEODORE E. BRIDGE, a mechanical engineer with the University of Chicago in connection with a project under construction for the Argonne National Laboratory, acquired a wide experience in the design of power and of process piping during 15 years with E. I. du Pont de Nemours & Co.; he has also had experience in the field of very high pressures. ***In his article, of which this is the first part, his objective has been to help the "average engineer" in applying theory to the problem of designing piping systems for conveying flashing mixtures of steam and water. His article is, we feel, a significant and most practical contribution to the subject of piping***

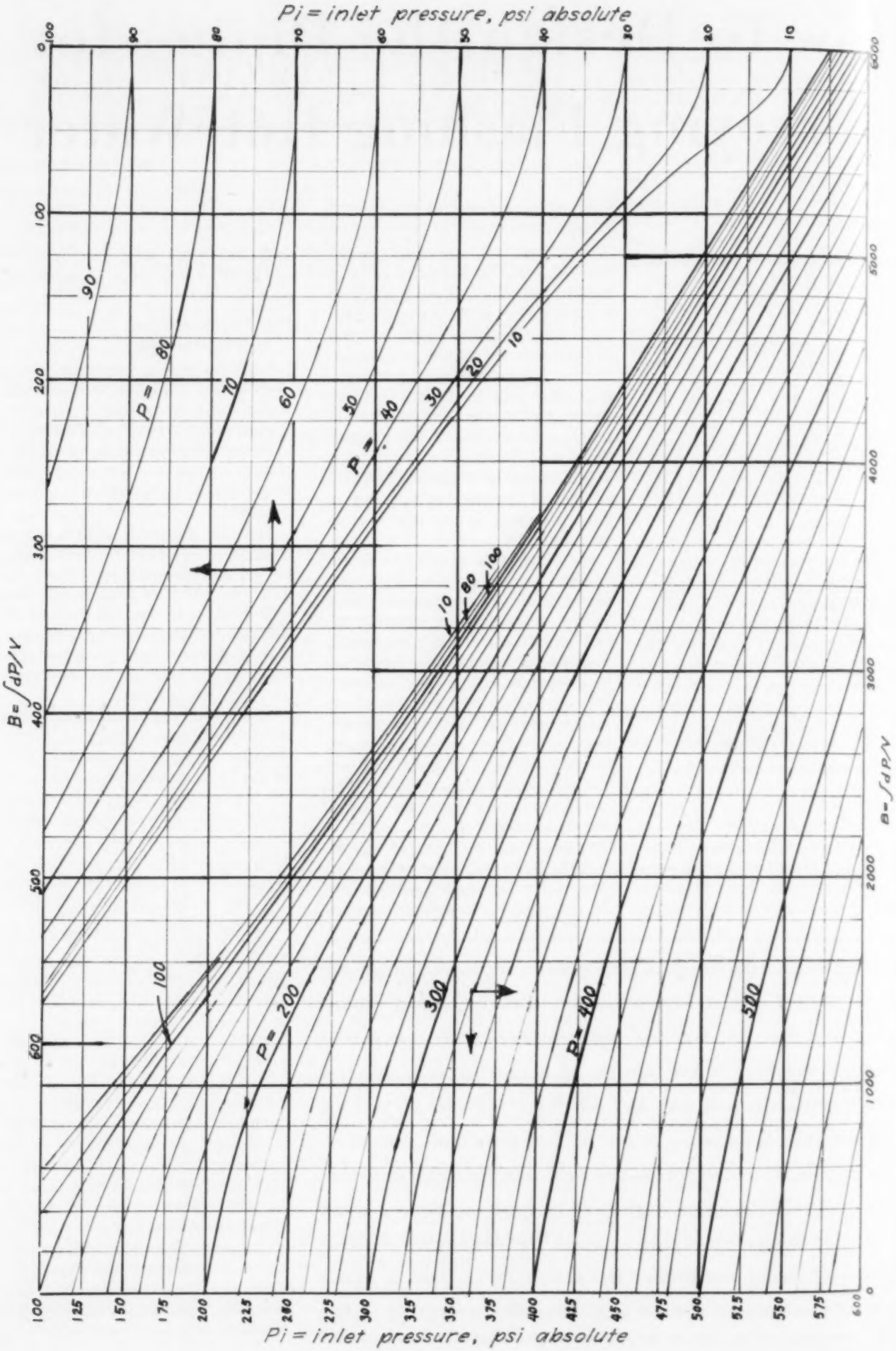

Pi = inlet pressure, psi absolute
B = ∫dP/V
B = ∫dP/V
P = 60
70
80
90
P = 40
50
60
30
20
10
P = 200
100
P = 400
300
500
Pi = inlet pressure, psi absolute

Fig. 1 (Opposite)—These curves are used to determine the value of the integral $B = \int dP/V$ when employing Equation 7 in Appendix A for calculating pressure drop in a pipe line. (dP is total differential pressure drop, psi; V is the specific volume of the steam-water mixture in the pipe, cu ft per lb; P is the pressure at any point in the system, psi absolute)

may have been caused by overdesign. The author has introduced empirical methods for quickly designing a condensate system to obtain a reasonable pressure drop in both halves of the system—namely, the orifice and the piping.

Orifice in a System Complicates Problem

There are several difficulties which will be encountered when one tries to calculate the pressure drop in a piping system conveying flashing hot water. The pressure drop may be calculated by using Equation 7 as developed in Appendix A. It will be seen that, in using this equation, it is necessary to calculate the term B by very laborious methods. However, by the use of the curves in Fig. 1, this part of the task is greatly simplified, as precalculated values for this term are given. Even so, the problem is somewhat complicated by critical flow effects, which will be explained later.

As soon as an orifice is introduced into the system, the problem becomes even more complicated. Consider the problem again of a condensate system with a trap installed at the inlet end of a long line. The length of the line (L'') and its diameter (D) are given. The problem is to calculate the area of the trap orifice (Ao) that will give a reasonable flow through the system. It is also necessary to calculate the maximum flow (M) that the system will pass. It will be shown later that the outlet pressure of the system must be equal to or greater than the critical pressure (P'') which may be read from the curves in Fig. 7. One next assumes what appears to be a reasonable value for the trap outlet pressure (Po), and then his troubles really start. With all of these values fixed, it is possible to calculate the equivalent length of the trap orifice (Lo) by using Equation 7. But this can only be done if one assumes a value for M, the flow through the line. The calculated value of Lo, when added to the actual length of the pipe in the system (L'') will give a value for T, the total equivalent length of the system. Using this value of T in Equation 7a, one can solve for M to see if his original assumption was correct. This cut and try process is very laborious and therefore the author has introduced some empirical curves which will yield a reasonable solution on the first try.

Calculating Surge Pressures

In addition to calculating the pressure drop of a piping system, it is also very desirable to calculate surge pressures resulting from high velocity of condensate around an elbow in the system. Normally, it would be quite a problem just to calculate the velocity of the steam-water mixture through the pipe, because of the difficulty in figuring the amount of flashing that has taken place at any given point in the system.

A method is presented in Appendix C which greatly simplifies this process by using precalculated data given in Fig. 2. After figuring the velocity, the surge may be calculated by using Equation 15, which is also developed in Appendix C. This equation gives a basis for calculating the radius of the pipe bends which are to be installed in the line. It is recommended that long radius bends be installed wherever the surge pressure is figured to be high. Also the pipe line must be anchored to withstand the unbalanced thrust of SA, the surge pressure times the cross sectional area.

After reviewing the theory developed in Appendix A, Equation 7

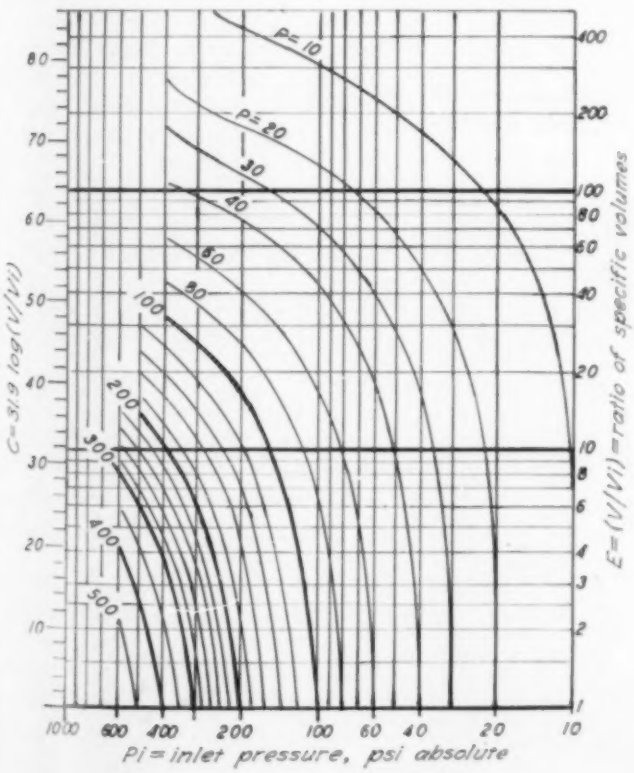
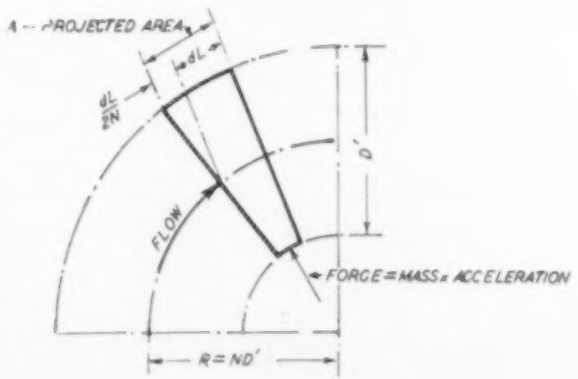


Fig. 2 (Left)—These curves are used to determine values for the constants C and E which are needed in Equation 7 when calculating pressure drop and Equation 11 when figuring velocity, respectively. (V is specific volume of the steam-water mixture in the pipe, cu ft per lb; Vi is the specific volume at the inlet of the system). Fig. 3 (Below)—Inertia force acting on a slug of water traveling around an elbow at high velocity (dL is differential length of pipe, ft; N is radius of pipe bend expressed as number of pipe diameters)



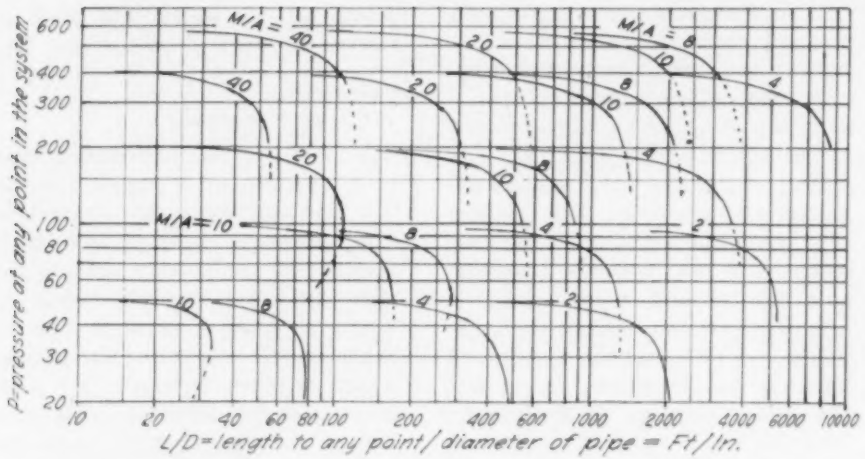


Fig. 4—Results of 21 separate calculations showing how the pressure in the pipe varies with the length (*M* is mass flow through the system in thousands of lb per hr, and *A* is cross sectional area, sq in.)

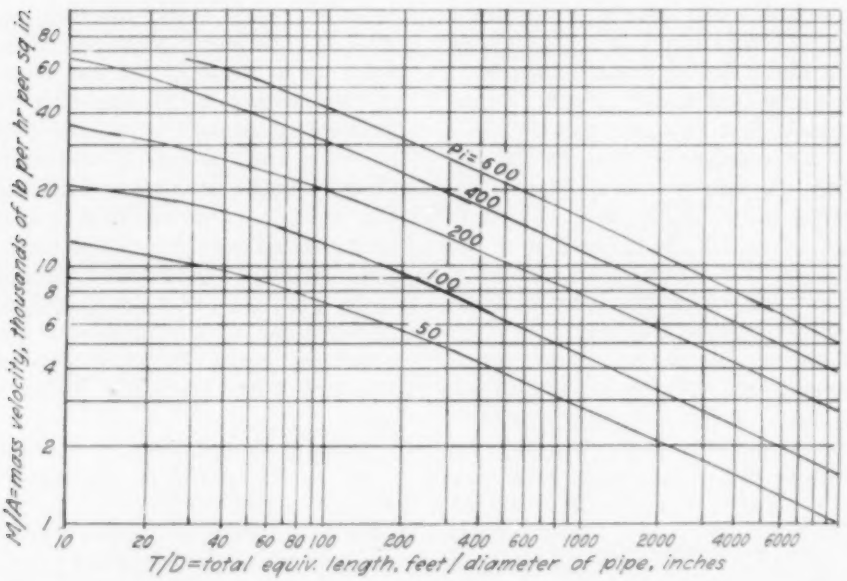


Fig. 5—How the mass velocity through the system varies with pressure and the equivalent length of the system (*Pi* is inlet pressure, lb per sq in. absolute)

was studied to determine how precalculated data could be presented to aid in its solution. As a result, the curves shown in Figs. 1 and 2 were calculated and plotted as shown. Next, 21 separate problems were calculated and the results plotted as shown in Fig. 4 to study the effect of critical flow in a piping system. If one of the curves in Fig. 4 is studied in detail, it will be noted that, for any given mass velocity (M/A), the pressure (P) falls very slowly, for a time, as the length (L/D) is increased. As L/D is further increased, the slope of the curve becomes steeper until it finally becomes vertical. The point at which the curve becomes vertical indicates the critical pressure. In effect, it shows that no matter how much more pressure drop is taken in the line, the flow through the line cannot be increased. That is what is meant by critical pressure. The value of L/D at which the line becomes vertical is called the critical length of the line.

In solving actual problems, this critical length of line is taken as equal to T whenever the pressure in the flash tank (Pf) is less than the critical pressure (P''). Since it is quite common for the pressure in the flash tank at the outlet end of a piping system to be less than the critical pressure, it is common for critical conditions to exist in the piping system. In such cases, the velocity at the end of the line will be very high. If the pipe were made very very large, it becomes, in effect, a long flash tank; and the velocity in such a pipe might be less than critical. For all practical purposes, however, it is not possible to reduce velocity by making the diameter of the pipe larger.

The data presented in Fig. 4 could be used as is, if it were not for the difficulty of interpolating for intermediate values between the curves. For this reason, the data in these curves were replotted in Fig. 5 for easy interpolation. [The Tube Turns Catalog and Engineering Data Book, No. 111, published by Tube Turns, Inc., Louisville, Ky., gives a group of curves which agree well with those presented in Fig. 5, thus giving an independent check on the method.]

When an orifice is introduced into the problem, it is not enough to know the maximum flow that can be forced through a given line; it is also necessary to know the shape of the P vs. L/D curve. This is necessary so that one may assign a fair proportion of the total pressure drop to each part of the system, namely the orifice and the piping. If the shape of the curve is known, it is possible very quickly to deter-

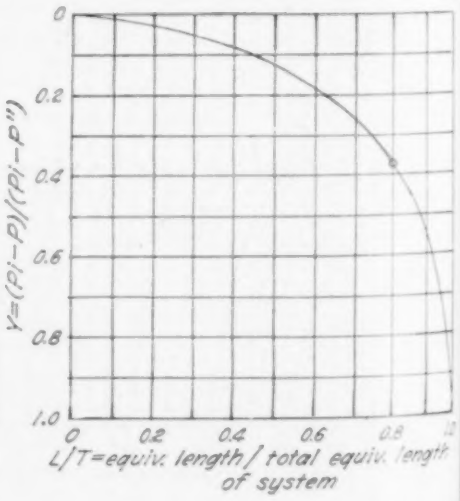


Fig. 6—This curve represents approximately any one of the curves shown in Fig. 4. It is accurate at only one point as indicated by the small circle on the curve. (*Pi* is pressure at inlet of system, psi absolute; *P* is pressure at any point in the system, psi absolute; and *P''* is the critical pressure as read from Fig. 7, psi absolute)

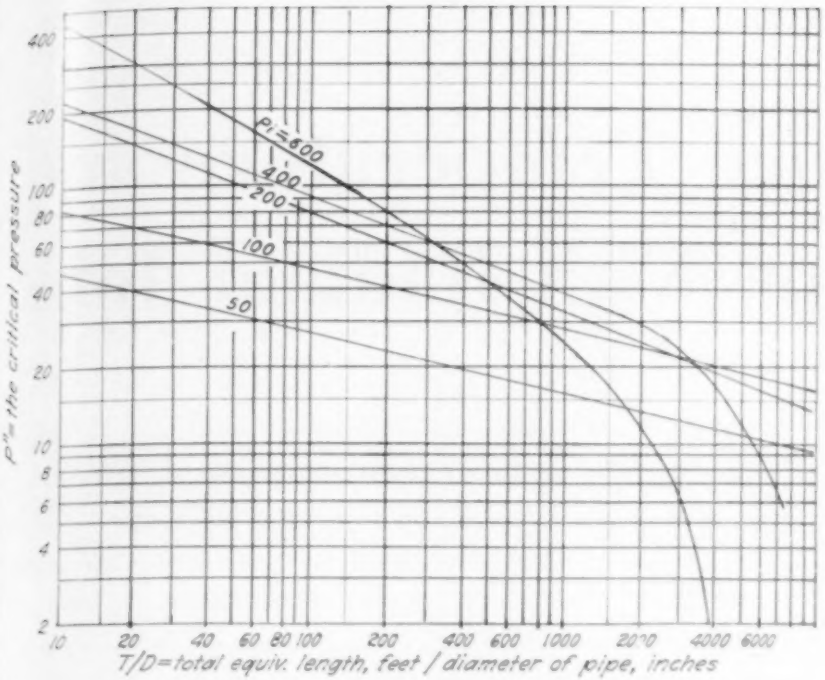


Fig. 7—These curves are used to determine the critical pressure in the system, *Pi* being the pressure at the inlet of the system, psi absolute

mine *Lo*, the equivalent length of the orifice. In order to be able to predict the shape of any curve, for all values of mass velocity (*M*/*A*) intermediate to those plotted in Fig. 4, a single curve, which approximately represents any one of the 21 curves in Fig. 4, was plotted in Fig. 6 against dimensionless coefficients. It should be noted that the curve plotted in Fig. 6 is accurate at only one point, and then only when it is used in conjunction with the curves in Fig. 7 which determine critical pressure.

The data plotted in Fig. 7 were deliberately distorted to be used in conjunction with Fig. 6 and yield accurate results at the one point indicated on that curve. This means that the curves shown in Fig. 7 will not give strictly correct values of critical pressure, but nevertheless the values as given will not introduce an appreciable error into the answer.

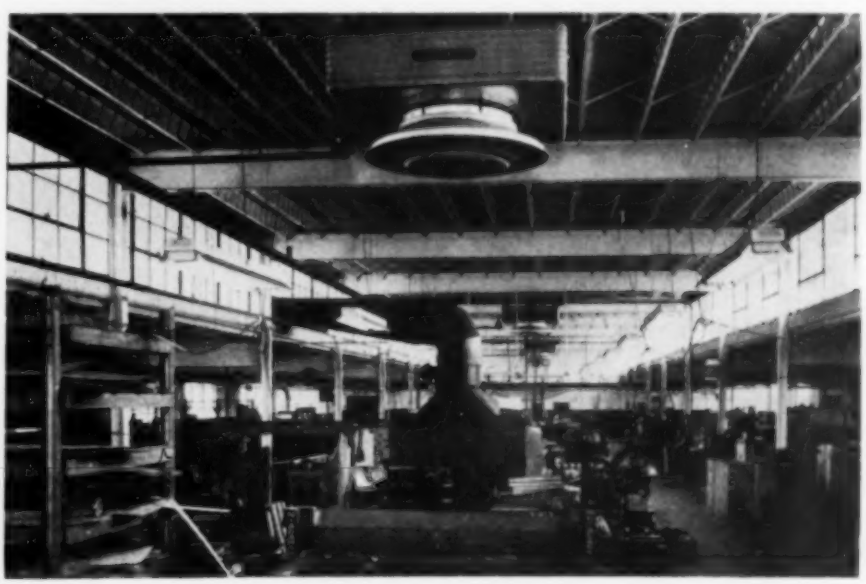
This is quite apparent when one studies the shape of the curves in Fig. 4. It can readily be seen that a considerable error in a value chosen for critical pressure (P'') will make only a slight change in the corresponding value read from the curve for L/D. This is evident from the steepness of the curves. This approximation, made in the construction of the curves plotted in Fig. 7, greatly simplifies the solution of problems involving an orifice in a piping system.

The sample problems to be presented in the next installment will illustrate the application of this method.

[To be continued]

Low Mounting Height for Big Unit Heaters

By installing air diffusers on the big vertical unit heaters for a Peoria, Ill., factory, they handle 11,000 cfm at 2200 fpm without discomfort to the employees, although the mounting height is but 13 ft. (Photo courtesy W. B. Connor Engineering Corp.)



THE original plans for an addition to a Peoria, Ill., factory called for a high monitor roof and vertical heating units were selected to deliver 11,000 cfm at 2200 fpm. Later, it was decided to reduce greatly monitor and roof height.

When the order for the unit heaters was entered, the change in the roof design was overlooked. Instead of being able to mount the units 20 ft from the floor, the maximum mounting height was found to be 13 ft and it appeared that such a low height for units of this size would prevent employees from working within a 30 ft radius.

However, by installing air diffusers on the unit heaters, the air stream was projected horizontally, covering a 75 ft radius, drafts eliminated to employee satisfaction, and what is believed to be a world's record established for low mounting of vertical heaters of this size.

Floor Surface Temperature and Heat Output With Radiant Jobs

PAUL S. PARK, manager of the engineering service dept., A. M. Byers Co., Pittsburgh, explains the disagreement between theoretical limits of 30 to 40 Btu per hr per sq ft of floor for radiant heating systems and the success being obtained with installations calculated to produce 60 to 100 Btu. It's a live subject these days, and his discussion includes points pertaining to the fundamentals of design of all types of heating systems

One of the most controversial questions on the subject of radiant heating is: "How much heat can be obtained from a floor type radiant heating system without producing uncomfortable floor surface temperatures?" For many years the application of some basic laws of heat transfer has created the impression that under no conditions should a floor type system be designed for calculated heat loads above 40 Btu per hr per square foot of floor surface; some handbooks have even limited the calculated output to 30 Btu, on the premise that higher outputs would produce excessive surface temperature. These figures have been arrived at by careful mathematical analysis of heat transfer by radiation and "natural" convection from a heated surface and—in view of the lack of evidence to the contrary—were widely accepted in this country as well as in England.

The controversy arises, however, when we examine the records of hundreds of actual installations of all kinds and find highly satisfactory performance and complete comfort being provided by floor type radiant heating systems which are required by standard heat loss calculations to produce 60, 80, and even 100 Btu per hr per sq ft of floor area. Many individuals (some of whom are capable and reputable engineers and scientists) have branded these figures as "fantastic," "based on errors in calculations," "products of imagination," and "impossible." Presumably, these remarks were made in good faith and on the basis of knowledge gained by study and, in some cases, experience. Whether right or wrong, their theories, experimental studies, and calculations have been published in scores of magazine articles, handbooks, and textbooks and have been much more readily adopted than the empirical data which have been developed from field investigations.

The unalterable fact remains, however, that theory and practice do not agree and the purpose of this discussion is to analyze the differences between the two and attempt to arrive at some logical conclusions.

Since the proponents of the theo-

Heating requirements of St. Nicholas church, Struthers, Ohio, range from 53 to 90 Btu per hr per sq ft of floor area. Arsene Rousseau, the architect and designer, states that the floors have been perfectly comfortable during the two heating seasons the system has been in operation

retical approach to the problem must necessarily depend upon basic principles of heat transfer, it is essential that we first study the means by which heat can escape from the floor and, if possible, discover some source of output which has been overlooked. Secondly, since the practical values are so much greater than the theoretical figures, we should carefully scrutinize not only the accuracy of conventional heat loss calculation procedures but the adaptability of such procedures to radiant heating.

For the sake of clarity, these two basic considerations can be broken down into the following four factors, each of which will be discussed individually:

1) Errors in the theoretical analysis of heat transfer.

2) Differences between radiant and other heating systems.

3) Margins of safety in standard heat loss calculation coefficients and procedures.

4) Safety factors introduced by individual designers.

Errors in Analysis of Heat Transfer

Not being a scientist, I have always been hesitant to question the theoretical procedures which have been developed by experts in the fields of mathematics, physics, etc. However, a close study of the published data available indicates that there is one potentially important source of heat transfer which may have been overlooked—or, at least, incorrectly evaluated.

Heat is released from a warm floor in a radiant heated space by two basic processes—radiation and convection. Radiation emission seems to be the simpler of the two to analyze because the fundamental formulas of Stefan and Boltzmann regarding radiant heat transfer between plane surfaces can be adapted to the heat exchange problem in an enclosed room. While the irregularity in room surfaces and materials may introduce some error in the practical application of radiation formulas, all authorities with whom I have discussed this phase of the subject are in agreement that the heat transfer by radiation can be considered accurate within 10 percent. Some test work we have conducted along this line substantiates their opinion.

Convection

Heat transfer by convection does not seem quite so easily determined and may contain a factor which is partly responsible for the wide variations between theory and practice. Here again, mathematicians and physicists have relied upon accepted basic formulas of scientists like Nusselt and Hencky, Hottinger, and others, and while we are not in a position to question the accuracy of any fundamental formula which has been proved over and over again by scientific laboratory experimentation, we are inclined to question the method of their application to the radiant heating problem. As it is so clearly pointed out in one handbook I am familiar with, these formulas apply only to the heat transfer resulting from "natural" convection created by the temperature difference between the room air and the heated surface. In other words, the convective output calculated on this basis represents only that heat removed from the floor by recirculation of 70 F room air and includes no allowance for heat removed by infiltrating cold air or the downward movement of cold air from large glass areas or other low temperature surfaces.

The magnitude of such an error of omission will vary with each different room, but it is common to find that infiltration losses alone represent 25 percent or more of the total heat load. If we are correct in our analysis up to this point, then it logically follows that the heat output theoretically possible by means of radiation and "natural" convection must be increased by an amount equal to all (or, at least, a major part of) the infiltration loss.

What allowance should be made for downdraft air from fixed glass and similar surfaces is not known. While such air movement is difficult to detect in a radiant heated space, physical laws demand that it take place and thus that portion of a heated floor adjacent to a large expanse of glass will have contact with air at a temperature below that of the room air and will lose heat by convection at a faster rate than published theoretical data indicate. Many automobile showrooms having glass areas covering one-half to three-fourths of the total exposed wall are being satisfactorily and comfortably heated in below-zero weather by a floor type radiant heating system. Heat losses for areas of this kind normally range between 70 and 110 Btu per hr per sq ft of floor and it is our opinion that the added convection output induced by the cold glass surfaces is at least partly responsible for the successful performance of such floor systems.

The theory has been advanced that if such convection currents are present, they will have a cooling effect on the floor surface and depress the surface temperature below the desired level—resulting in a reduction in heat output by radiation and "natural" convection. In my opinion, such a conclusion is the result of an incomplete analysis of the heat transfer problem. Naturally, the movement of cold air across a warm surface will tend to decrease the temperature of the surface and will do so if the source of heat which is warming the floor cannot be varied to meet changing conditions. But in a radiant heating system we have a variable source of heat and if the surface temperature of the floor is depressed to the point where the room is no longer comfortable, then the automatic controls will energize the circulator, mixing valve, or boiler firing equipment—thereby providing the additional heat needed to maintain surface temperature. From observation of several hundred actual installations, it appears that this entire cycle of heat demand and heat output becomes automatically balanced and with proper controls remains so throughout the heating season.

It is also conceivable that the net effect of air movement is to reduce the floor temperature from a 90 or 95 F level to within the more desirable range of 85 to 90 F. For example, a room having no infiltration loss might theoretically require a 92 F average floor surface temperature to offset the conductance losses through walls and roof, whereas an identical room having 25 percent of the same total heat loss attributable to infiltration will function satisfactorily with an average of 85 F as a result of the cooling action of the infiltrating air. This example is, of course, strictly a hypothetical case used to

Average heat loss in patients' rooms in the one story, basementless Montgomery county hospital, Dayton, Ohio, is calculated to be 67 Btu per hr per sq ft of floor area. Rial T. Parrish, architect for the structure, reports that the radiant heating system is completely satisfactory and that floor surface temperatures have never been excessive during the 2½ years the system has been in operation

illustrate the point under discussion.

By considering the problem as a simple type of heat exchange, it can be seen that the concrete surrounding the heating elements is merely the heat exchanger wall through which heat is passing from a hot fluid to a cooler fluid. The quantity of convected heat that flows is a function of the temperature differential between the two fluids and the surface temperature of the exchanger wall is incidental. If the temperature of the hot fluid can be held constant as the temperature of the cooler fluid is decreased, the surface temperature of the exchanger wall will decrease also, even though the rate of heat transfer by convection is greater. Heat exchanger output may also increase with velocity and without increasing the temperature of the exchanger surface, although no publications I am familiar with have evaluated or included this factor. This approach to the heat output question appears complex, to be sure, and while I am not prepared to specify the theoretical limit in every individual case, I do not believe a possible source of heat output should be ignored simply because we cannot yet accurately evaluate it.

Skin Temperature

In studying theoretical data, I have also noticed that a maximum floor surface temperature of 85 F has been arbitrarily set as the upward limit for comfort. One notable exception is the suggestion of Dr. F. E. Giesecke, in his book on Hot Water Heating and Radiant Heating and Radiant Cooling, that the safe limit for floor surface temperature be raised to 90 F.

Editions of the American Society of Heating and Ventilating Engineers Guide sometime previous to 1947 contained data estimating the mean skin surface temperature of a clothed adult at 83 F. The 1947 and 1948 editions of the Guide have raised this figure to 90-93 F and yet no corresponding change in allowable floor surface temperature has been discussed. If some theoretical relation existed between mean skin temperature and floor temperature when 85 F was selected as a safe limit for floor surfaces, then some upward adjustment seems logical if the mean skin temperature has been revised 7-10 F. Admittedly, 85 F is a safe maximum, but if 88 F or 90 F is also within safe limits, we can substantially extend the safe output range of floor type radiant heating systems and perhaps take another step toward closing the gap between theory and practice.

Effect of Uniform Air Temperature

It has been established that air temperature is quite uniform in a radiant heated space. Between floor and ceiling levels, readings with radiant heating will normally show only a 2 deg difference. The effect of a wide variation is to increase wall and ceiling losses appreciably above those present with radiant heating, and if systems can be satisfactorily designed under such conditions, we have no alternative but to assume that the use of identical procedures for designing radiant heating systems must necessarily introduce a margin of error in favor of radiant heating.

This error is intensified by the fact that comfort can be maintained with radiant heating at low room air temperatures. This does not mean thermostat settings in the 50's or the low 60's, but a more reasonable range of 68 to 72 F. The net result is to reduce heat losses and the error involved in using identical heat loss calculation procedures is again in favor of radiant heating.

A third (and perhaps the most important factor) is the effect of air movement in a room and the resulting temperature differentials between inside and outside air. These are quite important because heat loss from a structure varies directly as temperature differential and room air velocity, an increase in either or both resulting in an increase in heat loss. Air currents are created almost entirely by differences in the density of air at different temperatures (except for movement created by fans, etc.). Therefore with a maximum air temperature of 85 to 90 F in a radiant heated room, the temperature differential and movement of air

must be only a fraction of that in a heated space where the temperature of some of the air may be as high as 175 F. Thus, if standard heat loss calculation coefficients are accurate, their use in the design of radiant heating systems which create low velocities and small temperature differentials introduces another margin of error favorable to radiant heating.

Errors in Calculating Heat Losses

Every engineer is grateful to the American Society of Heating and Ventilating Engineers for the annual publication of the Heating, Ventilating, Air Conditioning Guide, which can be used as a reliable source of information for the calculation of heat losses from a building. It is particularly gratifying to note that a Guide publication committee is constantly active in recommending additions or revisions as new data are uncovered. The ASHVE's research staff has also prepared numerous bulletins that give us assurance that the Guide is as accurate and up-to-date as is possible within practical limits. Thus, while the Guide is accepted as our "bible," the continuing research and investigation indicate the need for revision or verification of many factors.

For example, Clark M. Humphreys, senior engineer at the society's research laboratory in Cleveland, prepared about a year ago a detailed report on A New Method for Selecting Winter Design Temperatures. One of the conclusions was that "the daily mean temperature is a better indication of heating load than the daily minimum temperature. It is therefore a more logical basis for the determination of design temperature." Tabular data published with the report (which appeared in the ASHVE Journal Section of the January 1948 HPAC) indicate that the minimum design temperature being used today may be 10 F, or more, lower than can be expected to occur every year or remain for any appreciable length of time. Mr. Humphreys' contentions are well supported and are valid evidence that we are designing our heating systems for greater capacity than is required except in rare instances of short duration.

Thus, radiant heating systems calculated to provide heat at a higher rate than is theoretically possible may be called on to do so infrequently and for so short a period of time that the thermal capacity of the structure may be sufficient for such critical periods to pass unnoticed. The net effect of such a condition would be that, for each job, some portion of the calculated heat requirement has no real bearing on comfort and should perhaps be added to the allowable theoretical limit.

The accuracy of commonly used heat transmission coefficients for building materials and sections is also a constant research subject. The values in the Guide have served our purpose adequately for years and are today universally used with success. However, these values are average coefficients representing one or more sources of information and thus for a well constructed building would logically be conservative. Experiments on conductivity of materials are being carried on and a recent one on flat glass (see ASHVE Journal Section, December 1948 HPAC) indicated that under other than conditions of nocturnal radiation the *U* value of 1.13 for single glass may be high. T. Napier Adlam, in his book on Radiant Heating, also suggests that lower values for glass and other materials are possibly correct, although he speaks of them only in conjunction with radiant heating. The air change method of infiltration calculation is admittedly only an approximation and yet its use is quite common. These are all possibilities of error on the conservative side, which combined may represent a measurable percentage of the difference between theoretical and practical limits of radiant heating.

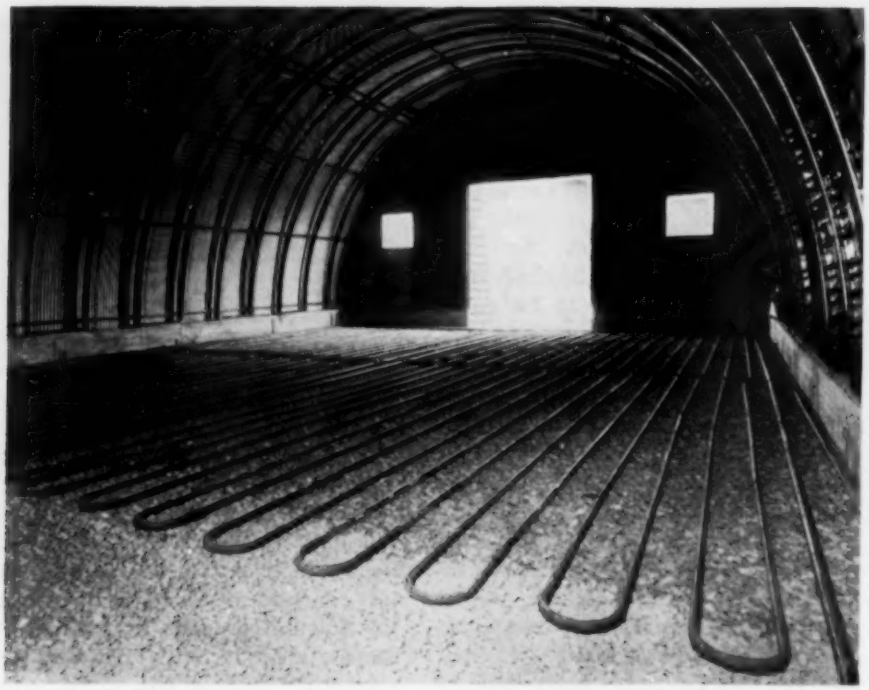
To the best of my knowledge, heating engineers do not make allowance for heat gain from occupants, mechanical processes, lights, and other electrical equipment. While these are variables and thus may never become a part of a standard calculation procedure, they are very real sources of heat which ought to be at least recognized when critical areas are being studied. Heat gain by itself may not represent much of a correction in a heat loss calculation, but when combined with the other margins of safety, it becomes part of a group of "straws," anyone of which may be the one "that breaks the camel's back."

Individual Designer's Safety Factors

It is quite reasonable to expect a designer to be conservative enough in his calculations to provide a margin of protection to cover extremely critical conditions or unforeseen changes of a minor character. It is my impression that this practice has been carried over to radiant heating to an even greater degree than with other types of systems. For example, a minimum coefficient of 0.10 for wall and roof sections may be used even though the calculated value may be less; the air change method of figuring infiltration may be followed whereas the crack method is considered more accurate for most buildings; areas, coefficients, and Btu losses are frequently rounded off to the closest even figure; exposure factors for north and west walls may be added; and, as a final adjustment, the total heat loss of the room or building may be increased by a flat 10 percent or so just to be "on the safe side."

The net result of including such factors is that the designer ends up with a fictitious heat requirement that is considerably greater than will ever be needed and well above the maximum theoretical limit recommended for radiant heating by some authorities. With radiant heating, the temperature of the heating surface is a critical factor and the use of unnecessary or excessive safety factors may result in a false requirement causing radiant heating to be ruled out or, if installed, to produce comfort at a calculated level which does not exist.

I do not mean to imply that the inclusion of safety factors by individual engineers is a major cause of the difference between theory and practice. On the contrary, it is probably a minor item—but thorough analysis of the overall problem seems to have revealed many minor items which collectively may represent an almost complete solution. At any rate, it seems highly inconsistent to establish theoretical limits by means of complex mathe-



The sheet steel, uninsulated building for the Chester Automotive Center, Inc., Chester, Pa., was equipped with a floor type radiant heating system. The owner required radiant heating even though heat losses were calculated to be 155 Btu per hr per sq ft of floor area. J. W. Meil, secretary-treasurer of the company, stated that during the severe winter of 1947-48 the results were completely satisfactory

matics and then rigidly apply those limits to some decidedly practical figures which may vary widely according to the routine procedure of the individual engineer.

Before presenting the conclusions which may be drawn from the foregoing analysis, it should be brought out that in no way is there any intention on the part of the author to discredit or criticize any individual or organization. On the contrary, the purpose is purely objective in an attempt to further the development of accurate, standardized data which can be used with assurance in the design of floor type radiant heating systems.

In Conclusion—the Problem Re-Stated

In conclusion, I should like to restate that the problem under discussion is that theory limits the output from floor type radiant heating systems to 30 or 40 Btu per hr per sq ft of floor area, whereas field investigations have revealed that floor systems which have operated successfully for one or more heating seasons are calculated to produce 60 to 100 Btu in the majority of cases surveyed. This difference between theory and practice has existed for at least five years and to the best of my knowledge no report dealing completely and fairly with both sides of the story has ever been published.

The primary purpose of this discussion is to present such an analysis of the problem and, while a complete solution is not to be expected from a simple study of this kind, it is hoped that it will explain why differences of opinion exist and perhaps result in some adequate and immediate field or laboratory research by some competent, unbiased organization. The ASHVE research program should successfully solve this problem, but the long-term character of this radiant heating investigation may not permit the release of useful data for several years.

On the basis of the factors discussed in the preceding paragraphs, the following are suggested as conclusions:

1) The maximum allowable output from a floor type radiant heating system should be revised to include, in addition to output by radiation and natural convection, an allowance for:
   a) Output resulting from infiltrating cold air.
   b) Output resulting from downdraft cold air.
   c) Output from increasing permissible floor surface temperature.
2) The calculation procedure for determining heating load for a radiant heating system should be revised, taking into consideration:
   a) Uniform room air temperature with radiant heating.
   b) Low room air temperature with radiant heating.
   c) Little air movement against cold surfaces with radiant heating.
3) Heat loss calculations should be made using the most accurate coefficients and factors available and on the basis of more practical minimum outdoor design conditions. If safety factors are employed for sizing heating equipment, the excessive heat lossses thus obtained should not be the basis for estimating floor surface temperature.

It will be readily admitted that data for evaluating each of these conclusions are not yet available and may not be for some time. Under these conditions, the establishing of a maximum output limit in each case is a matter for the judgment of the individual engineer. Fortunately, there are sufficient installations on record which have been designed and are operating without regard to theoretical limitations to enable exercising judgment and making a sound decision on the basis of experience. After all, if the "impossible" has been done, it can be done again.

SALES AND EARNINGS AT ALL-TIME HIGH

Carrier Corp., a leading manufacturer of air conditioning, refrigeration, and industrial heating equipment, reported a net profit of $2,710,412 or $4.19 per common share for the fiscal year ended October 31, 1948. This compared with $2,272,774 or $3.44 per common share for fiscal 1947.

The corporation completed sales in the amount of $54,456,394, and booked $52,676,015 of new orders. The comparable figures for fiscal 1947 were $52,910,590 and $47,686,249.

The earnings, sales, and new business figures constituted all-time high fiscal year records for the corporation.

The annual report of the corporation revealed that during November and December, the first two months of the present fiscal year, $8,045,167 of orders were booked, as compared with $8,308,587 in the corresponding period of fiscal 1948.

BASIC PRINCIPLES OF PIPING

(A Review of Fundamentals)

HERMAN VETTER, consulting engineer, Los Angeles, discusses in this third article of his series on fundamentals of piping the importance of proper pipe sizing, distribution header arrangement, provisions for drainage of condensate, and the avoidance of piping bottle-necks. He cites the case of a steam plant he recently visited where a 5 in. connection between the boiler drum and the header formed a bottle-neck and handicapped production.

IN THE STEAM generating plant water is converted into steam by the application of heat from fuel. In the refrigeration system, liquid refrigerant is converted into vapor in the process of absorbing heat from the air or from a substance required to be cooled. The process in either case is the evaporation of a liquid in the boiler or evaporator and condensation of the vapor back into a liquid so that it may be reused.

The difference between the two processes is in the temperature and pressure levels at which these operations take place. Water evaporates or changes into a vapor at 212 F when at atmospheric pressure, while ammonia liquid evaporates at —28.5 F at atmospheric pressure.

In order to discuss steam and refrigerant piping jointly, it will be necessary to use terms that apply to either system. As steam boilers are sometimes called evaporators and as the term evaporator is used to designate a similar piece of equipment in the refrigeration system, we will use here the term evaporator when discussing boilers. In the refrigerating system, the compressor occupies the same relative position as the engine in a steam plant; therefore the term engine will mean the engine in a steam plant or compressor in a refrigeration system.

In the analysis of vapor piping, it becomes necessary to understand what is meant by saturated vapor and superheated vapor; therefore, the following definitions are given:

Saturated vapor means vapor at the temperature corresponding to its pressure. For instance, steam at atmospheric pressure and 212 F is at saturation condition, commonly known as the boiling point. Steam at 100 psi absolute pressure and 327.81 F is also at saturation condition. If the temperature was lowered even a fraction of a degree without lowering its pressure, part of the steam would condense.

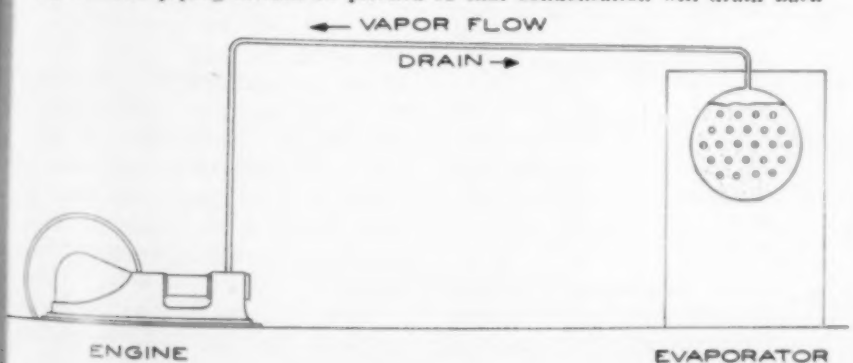
Superheated vapor is vapor whose temperature is higher than the saturation temperature (i.e., boiling point) of the co-existing pressure. Steam at 100 psi and with a temperature of 350 F is said to have 22.19 deg of superheat.

"Dry" saturated steam is steam free from mechanically mixed water particles. "Wet" saturated steam, on the other hand, contains water particles in suspension.

A wet saturated steam condition can result where steam is separating from water at high velocity, a condition that can exist in a steam boiler that is operating under an overloaded condition or one in which the steam connection from the boiler is too small, causing excessively high steam velocity.

Pressure drop is the difference in pressure between two points in a line; for instance, the difference in pressure between the boiler and the engine. A difference in pressure must exist in a line, otherwise there

Fig. 1—The piping should be pitched so that condensation will drain back



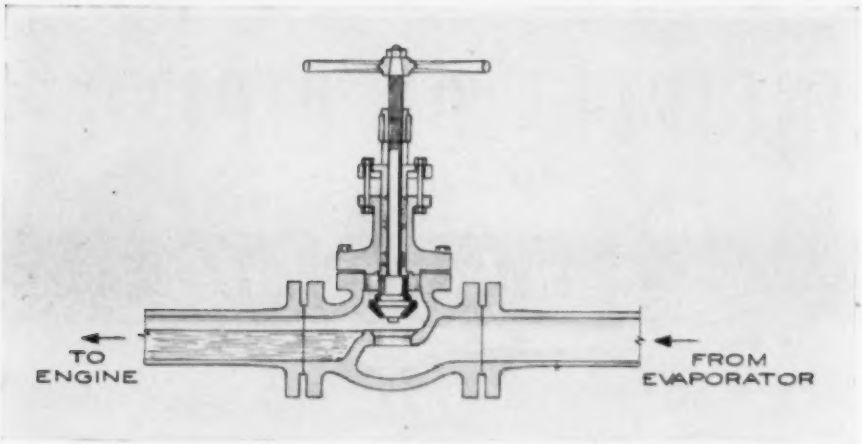


Fig. 2—Typical design of globe valve. If such a valve is in a horizontal line with the valve stem vertical, the side of the valve nearest the engine should be tapped and connected to a drain trap

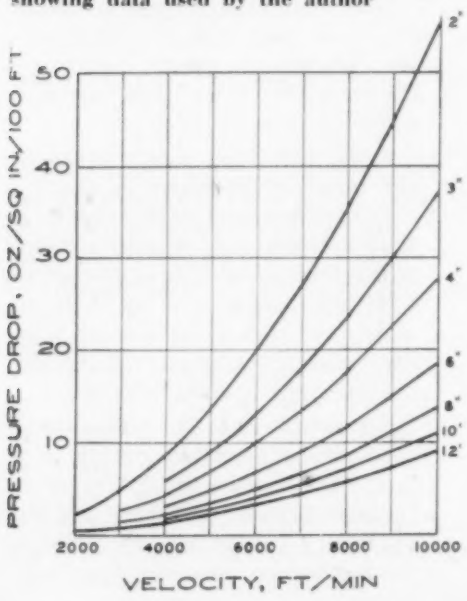
would be no flow through the line. Pressure drop is due to the velocity of the steam through the line and to friction of the steam against the wall of the pipe and to resistance caused by bends, valves, etc.

The piping between the evaporator and the engine should be pitched toward the evaporator, as indicated in Fig. 1, so that if any condensation occurs in the line the moisture will drain into the evaporator. This rule applies even where the vapor is highly superheated and the possibility of condensing moisture in the line during normal operation is therefore slight, for the reason that there are bound to be shutdown periods and when the piping is being reheated, condensate will collect at low points.

Typical design of globe and angle valves is shown in Fig. 2. In these valves, the seat is located at or slightly above the center line of the pipe, and when the valve is placed in a horizontal line with the valve stem vertical, the valve seat forms a dam behind which moisture can collect. When the valve is placed with the stem in a horizontal position, free drainage is permitted. If it should be necessary to place a valve in a horizontal line with the valve stem in a vertical position, the side of the valve nearest to the engine should be tapped and connected to a drip trap.

Pipe line sizes should be determined by the desired pressure drop and not by the velocity. Pipe line velocity is one of the factors used to determine the pressure drop, but velocity alone should not be the determining factor in sizing pipe lines. The size of the pipe as well as the length of run has a greater effect on the pressure drop than does the line velocity. All these factors are required to determine the pressure drop.

Fig. 3—Pressure drop in steam piping, showing data used by the author



There are several methods for calculating pressure drop in steam piping; some of these are rather complicated and require data on the quality of the steam that are not always available. The following is a simple equation used for air piping that can be used for general steam piping design:

$$p = \frac{Lv^2}{25{,}000\ d}$$

where p = loss of pressure, oz per sq in.; L = length of pipe, ft; v = velocity, fps; and d = inside diameter of pipe, in.

The pressure drop in pipe lines ranging from 2 to 12 in. diameter and for velocities from 2000 to 10,000 fpm is shown in Fig. 3. This indicates clearly that the size of the pipe has a great effect on the pressure drop and that the larger sized lines can be operated with higher velocities than the smaller sizes without too great a pressure drop.

Higher line velocities can be used for steam piping than for refrigerant piping—that is, if economy of operation is the deciding factor. There are conditions, however, where low mean steam velocity is desired and that is especially true where large quantities of steam are required on sudden demand. In such cases, high velocity should be avoided; otherwise the plant production may be retarded due to lag in steam supply.

The pressure drop due to valves and bends is expressed in equivalent length of straight pipe and is given in Table 1, for pipe sizes from 2 to 12 in. inclusive.

Table 1—Pressure drop due to valves and bends in equivalent length of straight pipe

Size of pipe	Globe valve	90 deg bend
2 in.	7	5
3 in.	13	9
4 in.	20	13
6 in.	36	24
8 in.	53	35
10 in.	70	46
12 in.	88	58

In Table 1, the resistance of valves and bends for the larger sizes of pipe seems to be out of proportion with that given for the smaller sizes. This is due to the fact that a greater area of turbulence is created in the larger size pipe from a change of direction and also because the pressure drop in the larger size pipe is less; therefore, the net difference in pressure drop is not as great as would be assumed by a comparison of the quantities given in Table 1.

Distribution Header Arrangement

The ideal header arrangement is one in which the pressure drop is the same to each engine. Fig. 4 shows an ideal header arrangement for connecting two engines and two evaporators. With this arrangement, the length of run and the number of bends is the same to either engine; therefore, the pressure drop to each engine is the same.

When, however, more than two engines and two evaporators are

connected with this same design of header, the piping becomes complicated, as can be visualized by referring to Fig. 5 which shows four engines and four evaporators so connected. The more engines and evaporators that are connected with this type of header, the greater is the complication.

The header arrangement shown in Fig. 6 is the one most commonly used. While the pressure drop to each engine will not be the same with this type of header, there are other advantages which outweigh the disadvantages.

Let us assume that we must design a plant for four 500 hp engines operating from four evaporators and that we wish to make a comparative analysis between these two types of distributing headers before making our decision.

Design Conditions for Example

Engines and evaporators spaced as shown in Figs. 5 and 6.

Steam consumption per engine: 20 lb per hr per hp x 500 hp = 10,000 lb per hr.

Steam pressure: 125 psi gage.

Engines and evaporators spaced 25 ft on centers.

Engines 100 ft from evaporators.

Steam Velocities

4 in. connection at evaporator: (10,000 lb per hr x 3.22 cu ft per lb) ÷ 60 min = 536 cfm x 11.312 = 6063 fpm, where 11.312 = reciprocal of area of 4 in. pipe in sq ft.

4 in. connection at engine same as above.

6 in. connecting yoke: 2 x 536 cfm x 4.984 = 5343 fpm.

8 in. main line: 4 x 536 cfm x 2.815 = 6035 fpm.

Pressure drop with header arrangement of Fig. 5

60 ft 4 in. (6000 fpm) = 0.6 x 10.00 (from Fig. 3) = 6.00 oz.

150 ft 6 in. (5000 fpm) = 1.5 x 5.88 (from Fig. 3) = 8.82 oz.

80 ft 8 in. (6000 fpm) = 0.8 x 5.00 (from Fig. 3) = 4.00 oz.

Total = 18.82 oz.

Pressure drop with header arrangement of Fig. 6

All piping except connections to engines and evaporators 8 in.

Steam velocity between engines 1 and 2: 536 cfm x 2.815 = 1500 fpm.

Steam velocity between engines 2 and 3: 1072 cfm x 2.815 = 3000 fpm.

Steam velocity between engines 3 and 4: 1608 cfm x 2.815 = 4500 fpm.

Steam velocity in main line same as in Fig. 5.

Item a: 170 ft 8 in. (6000 fpm) = 1.7 x 5.00 (from Fig. 3) = 8.50.

Item b: 50 ft 8 in. (4500 fpm) = 0.5 x 2.80 (from Fig. 3) = 1.40.

Item c: 50 ft 8 in. (3000 fpm) = 0.5 x 1.25 (from Fig. 3) = 0.63.

Item d: 50 ft 8 in. (1500 fpm) = 0.5 x 0.312 (from Fig. 3) = 0.15.

Item e: 10 ft 4 in. (6000 fpm) = 0.1 x 10.0 (from Fig. 3) = 1.00.

Total = 11.68 oz.

The pressure drop between evaporator 4 and engine 4 is

Item a: 8.50
Item e: 1.00

9.50 oz.

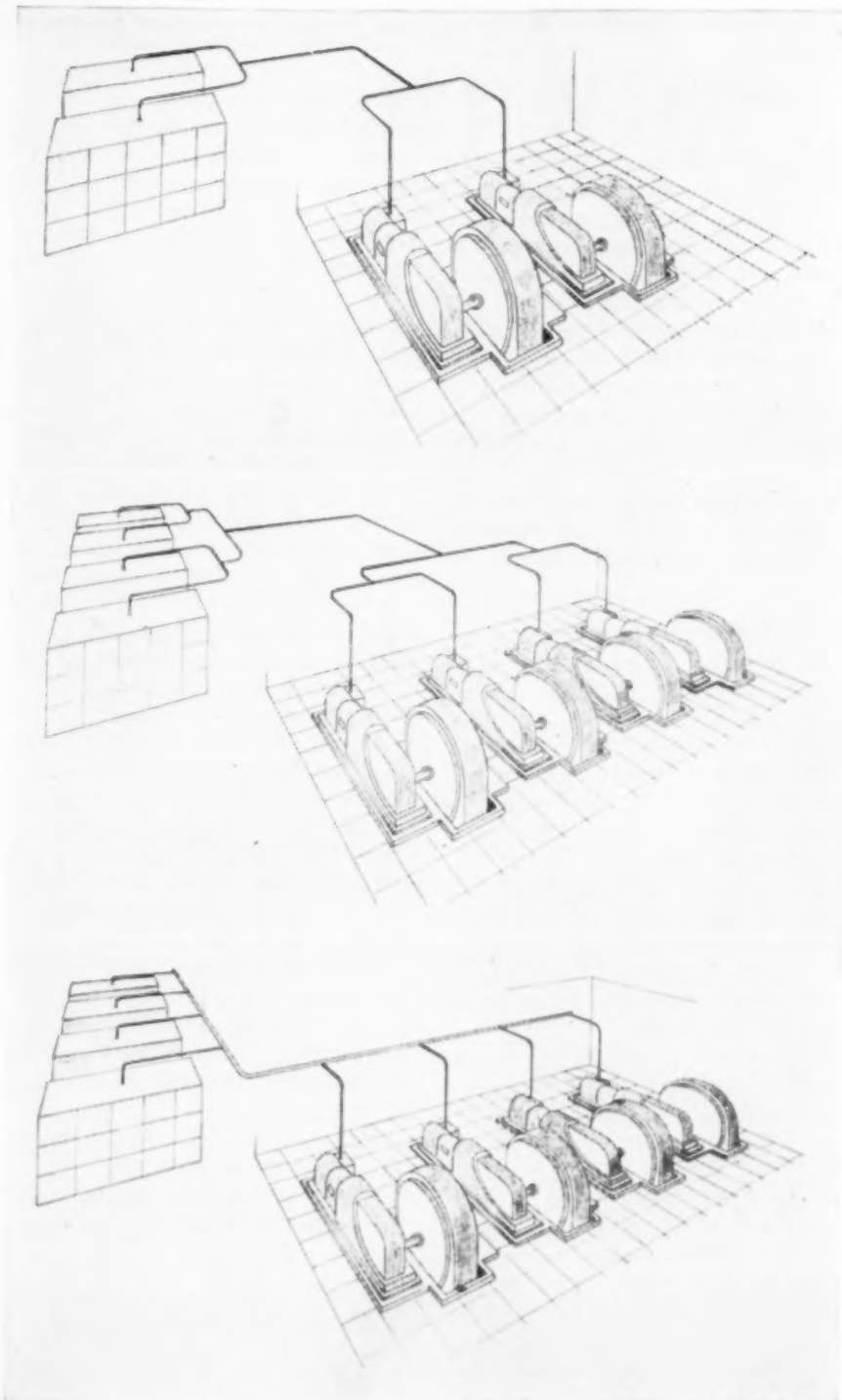


Fig. 4 (Top)—The ideal header arrangement is one in which the pressure drop is the same to each engine. Fig. 5 (Center)—When more than two engines and two evaporators are connected with the design of header shown in Fig. 4, the piping becomes complicated. Fig. 6 (Bottom)—This header arrangement is common

The average pressure drop therefore is

$$\frac{11.68 + 9.50}{2} = 10.59 \text{ oz.}$$

The pressure drop to each engine with the header arrangement shown in Fig. 5 is 18.82 oz, whereas the average pressure drop to the engines with the header arrangement shown in Fig. 6 is but 10.59 oz.

A difference in pressure drop in a steam header is not as objectionable as in a refrigerating system, because the governor on a steam engine will compensate for the difference in steam pressure whereas the refrigerating compressor has no means for compensating pressure differences. This will be discussed later.

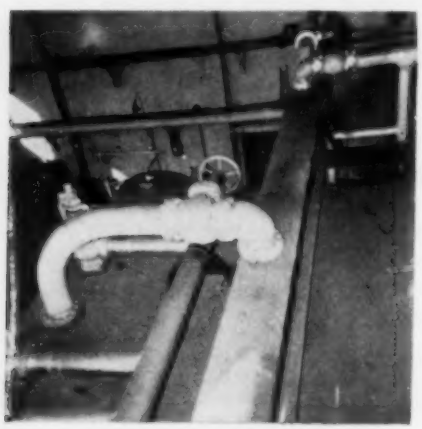
Avoid Bottle-Necks

In designing piping systems, care should be taken to avoid bottle-necks. A bottle-neck in a piping system will reduce the capacity of the piping system in the same manner as a stretch of narrow road in a major highway system will reduce the flow of traffic.

In a plant I recently visited, the chief engineer stated that when the steam demand was high, a lag in the steam supply often handicapped their production and necessitated overtime operation.

An analysis of the piping system brought out these facts: The connection from the boiler to the 12 in. distribution header is only 5 in. There are three 5 in. and one 4 in. supply lines from this header to various parts of the plant. One of the 5 in. lines supplies steam to a 650 hp engine driving an electric generator. The 4 in. line is for the water heater and auxiliary pumps. Two 5 in. lines supply steam for ironers, tumblers, and other machinery.

Fig. 7—In a plant recently visited by the author, the 5 in. connection between the steam drum and the distribution header formed a bottle-neck



It is quite evident that the 5 in. connection between the boiler and the distribution header is a bottleneck. Fig. 7 shows the 5 in. connection between the steam drum and the distribution header. This boiler has a 4 in. stub connection on the steam drum which could be used to provide a second connection between the boiler and the distribution header. This stub connection can be seen in the picture where the recess appears in the insulation near the end of the drum.

In analyzing steam piping problems of this kind, we must be familiar with the terms boiler horsepower and engine horsepower. Therefore, before discussing this plant further, let us define these terms. The following definitions of these terms are given:

Boiler Horsepower: One horsepower equals the evaporation of 30 lb per hr of water from an initial temperature of 100 F into steam at 70 psi gage pressure, which is equivalent to 34.5 lb per hr of water evaporated from a temperature of 212 F into steam at 212 F.

Engine Horsepower is found by the following equation:

$$\text{Hp} = \frac{PLAN}{33{,}000}$$

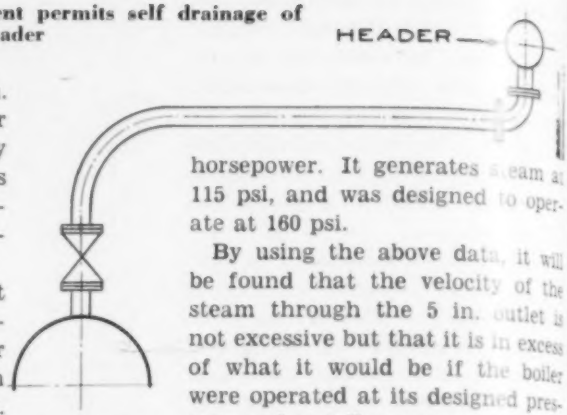
where P = mean effective pressure per square inch of the steam on the piston; L = length of stroke in feet; A = area of piston in square inches; and N = revolutions per minute.

The following data on lb per hr steam consumption of engines may be used where the definite value is not known.

Simple noncondensing	29- 45
Simple noncondensing automatic	26- 40
Simple noncondensing Corliss.	26- 35
Compound noncondensing	19- 28
Compound condensing	12- 22
Simple duplex steam pumps.	120-200
Turbines, noncondensing	28- 60
Turbines, condensing	12- 42

The boiler in the plant under discussion has a heating surface of 6225 sq ft and is rated at 650 boiler horsepower. It generates steam at 115 psi, and was designed to operate at 160 psi.

Fig. 8—This arrangement permits self drainage of condensate from the header



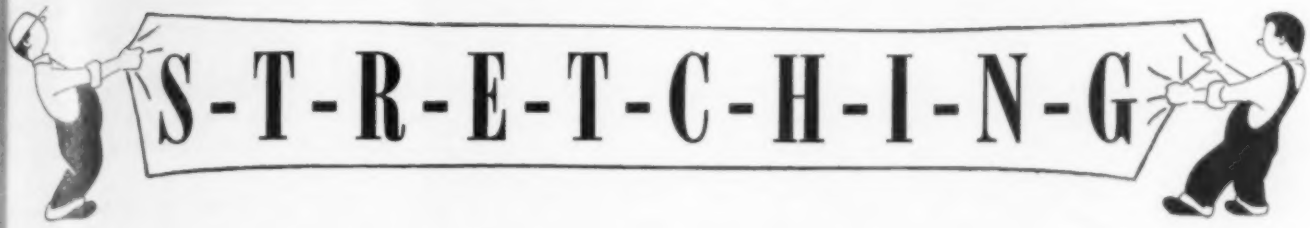
By using the above data, it will be found that the velocity of the steam through the 5 in. outlet is not excessive but that it is in excess of what it would be if the boiler were operated at its designed pressure and at full capacity.

The present operating conditions can be improved by connecting the 4 in. stub outlet, as the combined area of the 4 in. and 5 in. connections is slightly in excess of a 6 in. pipe.

While the arrangement of the distribution header is of common design, it will be noted by referring to Fig. 7 that the connection from the boiler enters the header at the top. This arrangement causes the header to become a moisture trap, preventing any condensate from returning to the boiler except by means of steam traps.

Comparing the arrangement of the steam piping shown in Fig. 1 with that in Fig. 7, it will be seen that in Fig. 1 the piping slopes towards the evaporator so that the condensate can drain back into the evaporator. The arrangement shown in Fig. 8 permits self drainage of condensate from the distribution header and connections back into the evaporator. For warming up purposes, a bypass valve is usually installed so that steam can be passed around the main stop valve so that the piping can be heated up more gradually than if the main valve were partly opened. This bypass valve also permits draining the header before opening the main stop valve.

In Fig. 8, the main stop valve is shown bolted to the outlet nozzle of the evaporator and not in the line between the evaporator and the header as shown in Fig. 7. In all pressure piping it is a good policy to locate the stop valves on the outlet of all pressure vessels so that there will be as few joints as possible between the stop valve and the source of pressure. In the event of leaks, the pressure can be shut off, permitting repairs without shutting down the evaporator.



Dehumidifier Performance

PETER FRANCK, of Armo Cooling & Ventilating Co., New York suggests a "run around" method, applied to the air cycle, for stretching the performance of a chemical or adsorption type dehumidifier. Part of the air leaving the unit is cooled and re-introduced into the unit, thereby lowering the moisture content of the entering air and—therefore—of the effluent air

FOR MANY air conditioning and refrigeration installations the use of a chemical dehumidifier and a source of cool water lead to economy in operation, low first cost, and accurate control of both temperature and humidity. However, the application engineer may find that the standard commercial dehumidifiers do not dry the air quite enough. He may try to "s-t-r-e-t-c-h" the performance of the unit either by using more air and a smaller moisture differential (thus increasing the ductwork, space requirements, etc.) or by choosing a larger dehumidifier so that the air velocity through the unit is decreased, which slightly improves its performance (and increases the cost).

It is for this "stretching" that the method described in this article is particularly useful—although it may also be applied in cases where a commercial unit is to be used to produce considerably lower humidities than indicated by its standard performance curve.

Run Around Scheme Utilized

Briefly, the method consists of the "run around" scheme shown in Fig. 1. A portion of the treated air after it leaves the unit is cooled and re-introduced into the unit, thus lowering the moisture content of the entering air and therefore also of the effluent air.

A difficulty arises in the computation of the performance of this cycle, since the re-introduction of the discharged air will continually lower the entering air conditions, and the question is, where will the performance stabilize itself?

Mathematically, this point is easily found as the limit of a converging sequence—provided the performance of the dehumidifier can be

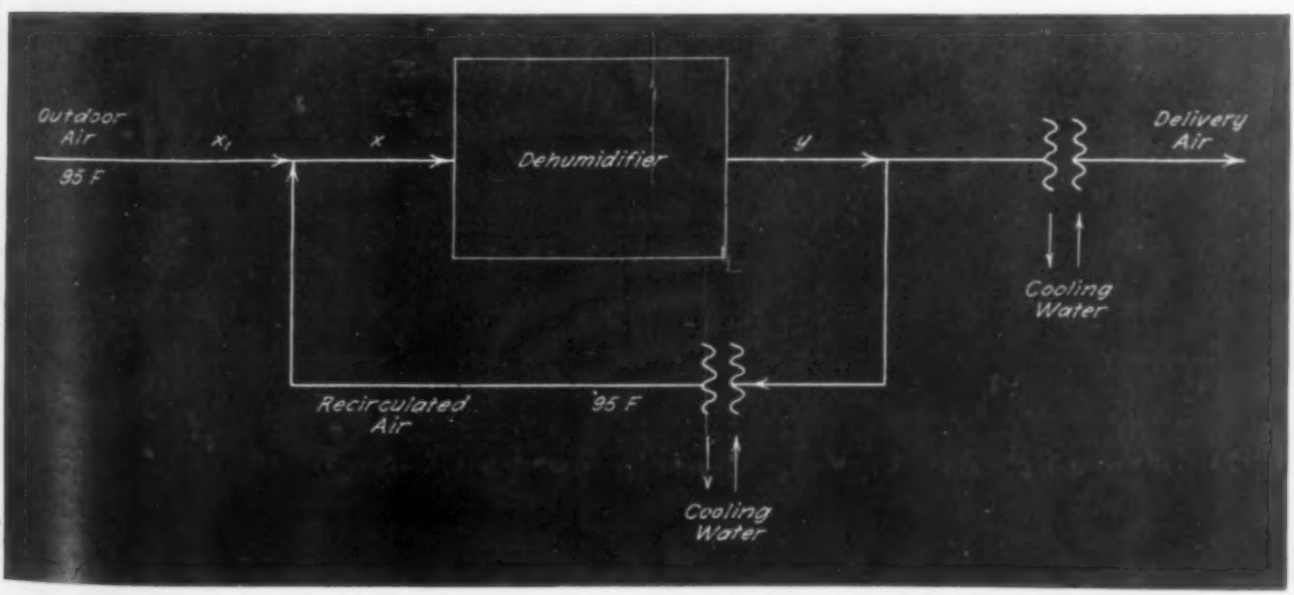


Fig. 1—The run around scheme diagrammed here may be used to stretch the performance of a chemical dehumidifier

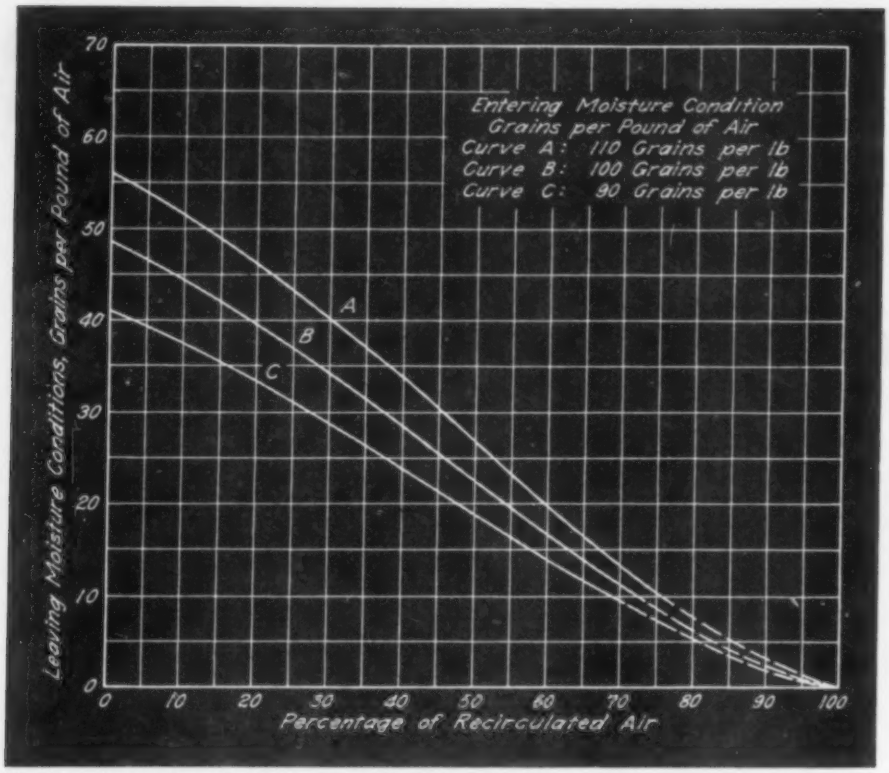


Fig. 2—A graph such as this permits determining the performance for various proportions of recirculated air

expressed in a mathematical function, as for example in the graph which is shown on page 682 of the 1948 edition of the ASHVE's Heating, Ventilating, Air Conditioning Guide. The translation of the graph into an analytic function is shown in Appendix 1 of this article, and the method of computation to establish the equilibrium point is shown in Appendix 2. From such computations, a graph can easily be constructed, as shown in Fig. 2, which permits determining the performance of the dehumidifier for various proportions of recirculated air.

Graph Determines Performance

The use of such a graph is self-evident. Along the curve of the given entering condition, we proceed to the intersection with the horizontal line indicating the desired leaving air condition and then find the percentage of recirculated air on the bottom scale. It must, of course, be kept in mind that only the balance of the air is available to be discharged into the room, but the loss of capacity of the unit is compensated for by the lower moisture content of the leaving air.

It is to be noted that the graph in Fig. 2 is only exactly valid for the temperature conditions indicated in Fig. 1; however, the performance curves in the Guide (referred to above) show that the influence of the entering air temperature is relatively slight.

Appendix 1—Translating Graph into Analytic Function

By plotting the performance curve as taken from the Guide (referred to above) on logarithmic paper it is seen that an exponential function would be a good approximation.

To permit a better approximation we use an equation of the form

$$y = a(e^{bx+cx^2} - 1) \qquad [1]$$

to fit to the performance curve. Here x is the entering moisture condition divided by 10 and y the leaving moisture condition divided by 10.

Table 1—Agreement between graph of performance and Equation 2

x	y	
	Value from graph	Computed Value
0	0	0
1	0.25	0.22
2	0.47	0.56
3	0.78	0.82
4	1.17	1.28
5	1.66	1.66
6	2.20	2.18
7	2.81	2.77
8	3.49	3.42
9	4.17	4.11
10	4.85	4.85
11	5.56	5.61
12	6.34	6.38
13	7.15	7.12
14	7.87	7.84

Equation 1 automatically satisfies the condition $x = 0$, $y = 0$; and the coefficients a, b, c are evaluated by substituting for x and y three values from the graph and then evaluating the three resulting equations for a, b, c.

By this process the author found the equation

$$y = 0.717(e^{0.275x - 0.077x^2} - 1) \qquad [2]$$

The agreement between the graph and Equation 2 is quite good, as can be seen from Table 1.

Appendix 2—Establishing Equilibrium Point

Let us follow a particle of air going around and around with the recirculated air. Its entering condition into the unit on its first passage is denoted by x_1; it is equal to the outdoor condition. Its leaving condition is given by Equation 1:

$$y_1 = a(e^{bx_1+cx_1^2} - 1)$$

with the appropriate values of a, b, c substituted, as in Equation 2.

Upon re-entering the dehumidifier, we have a mixture of the air at condition y_1 and x_1:

$$x_2 = a(e^{bx_1+cx_1^2} - 1)p + x_1(1-p) \qquad [3]$$

where p is the percentage of recirculated air.

This process continues so that the air entering for the nth time has the condition

$$x_n = a(e^{bx_{n-1}+cx_{n-1}^2} - 1)p + x_1(1-p)$$

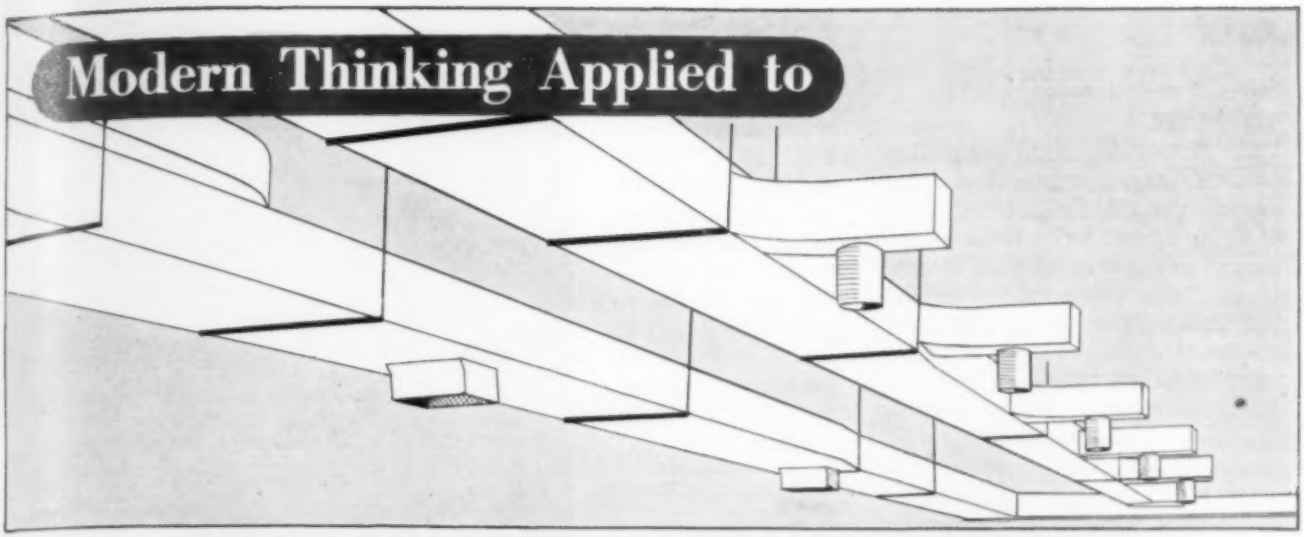
This is a recurrence formula, and we obtain, in the limit, representing the equilibrium condition,

$$x = a(e^{bx+cx^2} - 1)p + x_1(1-p) \qquad [4]$$

This equation determines x, the entering air at equilibrium conditions. It may easily be found as the abscissa of the zero value of the function

$$F(x) = x - ap(e^{bx+cx^2} - 1) - x_1(1-p) \qquad [5]$$

By computing the corresponding y from Equation 2, the final leaving air condition is found. Following this procedure, the points of the graph in Fig. 2 have been computed.



Modern Thinking Applied to DUCT DESIGN

KIRBY WALKER draws upon his experience both in the heating and ventilating field and in wartime research and engineering to discuss modern methods of duct design. A licensed professional engineer, he was engaged on heat transfer and combustion projects for two large aircraft companies during the war. He has had a wide experience in the heating and ventilating industry, and has supervised manufacturers' laboratory and development personnel. This is the first of a series of articles on modern duct design

PERHAPS IT would be well to set forth the reasoning that prompted my writing this article. Early in my career, I was responsible for designing and supervising many heating and ventilating systems. As in all heating and ventilating projects, lack of time and sometimes money forced the use of engineering shortcuts. These were rules of thumb sometimes based on bitter experience, sometimes taken from engineering texts based on the incomplete data of the time. Our theory was rough but served to produce workable systems.

During the past war, I was given the privilege of organizing and operating practical development and laboratory facilities that taught me a great deal about the true theory behind the subject of heat and fluid flow. I learned the practical use of nondimensional numbers and the application of dimensional analysis and the kind of modern thinking that makes possible things like gas turbines and jet propulsion. Looking now at the rule of thumb thinking which still exists in heating and ventilating, I see several spots where design thinking and practice could be assisted by a study of some of these relationships.

We work a great deal on paper without the chance to check our calculations in the laboratory. A ventilating system may work satisfactorily, but that doesn't give us an analysis of the various estimates we made of the action of the different parts and details of the system. We thus could go on for years in a "dream world" of our own, using and believing some fallacy—an incorrect understanding of some particular problem involved. Two errors could cancel each other out. By getting a correct understanding of the relationships involved, we may shorten procedure and produce better results.

Everything Comes from Test Data

First, it is important to realize that everything comes from test data. Our thinking is an attempt to explain what happened to produce the data in order that we may predict similar happenings in the future and thus make possible design procedure. We thus build up a theory, a plausible explanation of what actually happens in practice. Test data, the evidence, do not change as we learn. Theory changes. The practical, experienced engineer has a theory that is a plausible explanation of what actually happens. If someone says "this" happens in practice but "that" in the-

ory, the theory is wrong—not the test data.

Using Nondimensional Thinking

The second important realization necessary is an appreciation and use of nondimensional thinking. Many engineers have never really learned to think in nondimensional terms. To one who hasn't applied this thinking in practical design it appears confusing at first. But after learning to apply nondimensional thinking, a lot of things become simple mental operations that used to be complicated by involved arithmetic and frequent reference to engineering texts.

Percent is nondimensional. We all know how it helps in understanding certain figures. It would be considered ridiculous for two people to discuss the interest on a bank deposit as "I receive $10 interest." How much simpler it is to say "I receive 2 percent." The use of the nondimensional term "percent" paints a complete mental picture. In general, nondimensional terms are ratios which are the same value in any consistent system of numbers. The ratio must consist of two numbers in that system both raised to the same power.

This article is concerned with duct work carrying air at customary temperatures. The velocity head of the average velocity in a duct is herein termed q and is dealt with in inches of water column. Friction in a tube is simply a drag on the inner wall surface—just as friction on an airplane wing is evidenced by a drag on the outside surface of the airfoil. Mathematically, friction in a pipe and the friction about a wing are related.

In dealing with these matters, we note that this friction is always a fraction of q. When a duct designer says an entrance loss is ½ q, or an aircraft designer says the drag on a wing is ½ q, they are both talking the same language. Friction in an elbow, a piece of straight pipe, or any other unit of a duct system is therefore best expressed as some number times q. Feet of pipe or number of diameters for an obstruction complicates thinking and should not be used.

In connection with elbows and other obstacles to flow, it is not the purpose of this article to suggest methods for estimating the friction no matter what the design. It is, rather, to suggest simple rules by which simple practical parts for any situation may be designed and whose friction will be a minimum for the space involved and for which a friction value will be easily known.

Trial Computation

Another engineering attitude that is very helpful is a tendency to use trial computation. This consists of setting up several hypothetical situations, solving them, and comparing the answers. From that comparison the desired situation is picked. This method is opposed to that sometimes attempted—the setting up of a group of simultaneous equations to be solved for the perfect answer. This latter procedure is not practical for problems in heat and fluid flow.

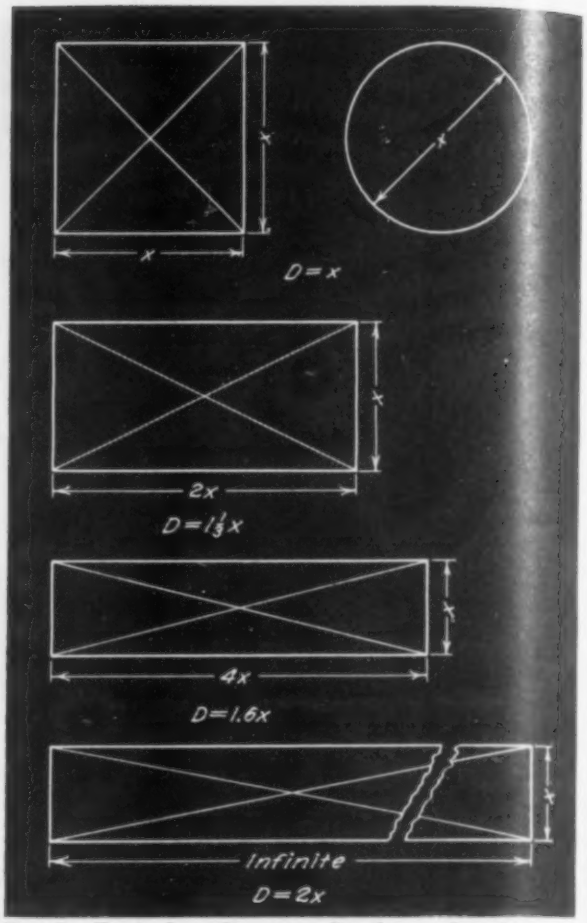


Fig. 1—Equivalent diameters for ducts of various proportions. The equivalent diameter, D, is always between once and twice the smallest dimension

"Lb per Hr" to Replace "Cfm"

There is a final step in modern duct design that will not be covered here but in my opinion will be adopted some years from now as procedure goes through the evolution brought about by increased knowledge. Ducts and outlets will be labeled in lb per hr and not in cfm. Mass velocity, lb per sq ft per hr, will replace fpm. Permissible velocities may be referred to as permissible q, velocity head. I have used this method to design wing and cabin heating on a wellknown airplane and can say from experience that it simplifies the work and produces a more efficient design.

Velocity Head —Quick Method

We must obviously have a quick way of knowing the q in order that we may intelligently follow design procedure. We have therefore:

$$q = \left(\frac{\text{fpm}}{\text{constant}}\right)^2$$

where q is velocity head in inches of water column of the average velocity; fpm is feet per minute, and the constant for air is

Altitude	0 F Air	70 F Air	140 F Air
Sea level	3700	4000	4300
5,000 ft	4100	4400	4700
10,000 ft	4500	4800	5100

The constant, 4000, is familiar to anyone using fans and the other constants given are close enough for practical purposes. For temperatures or altitudes between the figures tabulated, use rough interpolation.

Friction Coefficients and Reynolds Number

In the formulas, we use friction coefficients to express what part of q or how many q is a loss. These coefficients are roughly of two kinds. One of them, f, is that for straight pipe friction. For shock losses, those due largely to sharp changes in shape causing eddies and turbulence, we use a coefficient

equivalent diameter of a duct. For

the R.N. range we use. It is gener-

satisfactory for properly de[illegible]ned

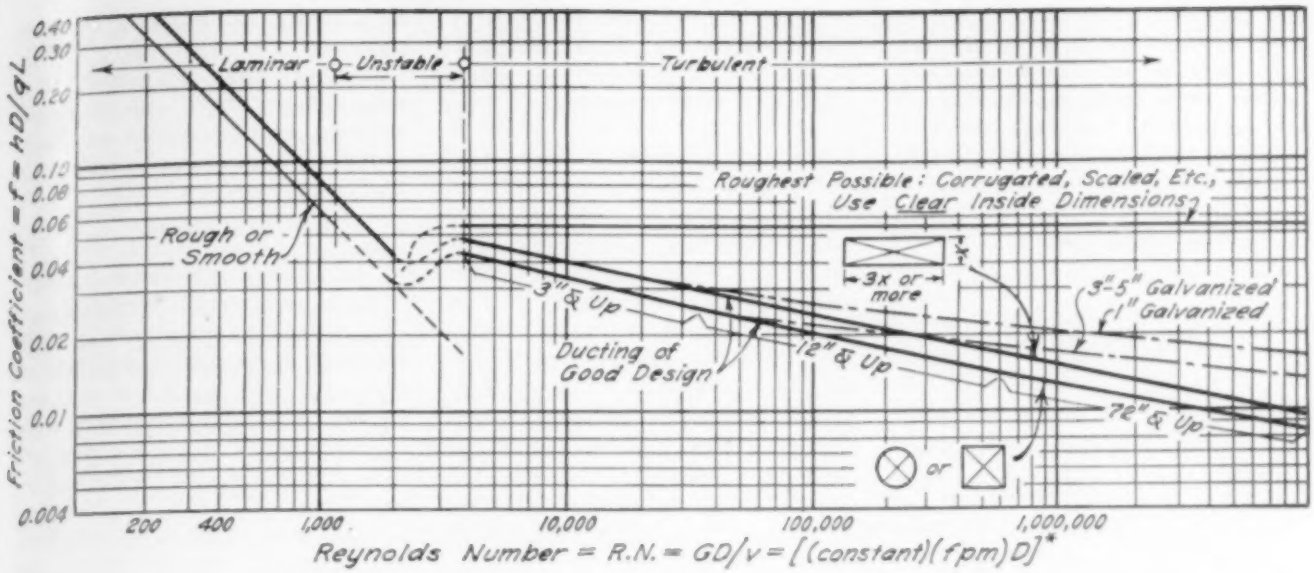


Fig. 2—Approximation of the results of hundreds of tests on round tubes showing friction factors for various Reynolds numbers

*Quick method—see text for constant.

$$D = \text{equivalent diameter} = \text{feet} = \begin{bmatrix}\text{clear inside}\\ \text{round tube}\\ \text{diameter}\end{bmatrix} = \frac{4A}{p} = \frac{1}{3}\begin{bmatrix}\text{sectional area, sq in.}\\ \hline \text{wetted perimeter, in.}\end{bmatrix}$$

SYMBOLS

A = area = square feet

cfm = cubic feet per minute

fpm = actual feet per minute velocity = cfm/A

G = mass velocity averaged across section = lb per hr per sq ft

h = loss of total pressure = inches of 32 F water column

L = length of straight duct = feet

p = wetted perimeter of duct = feet

q = one velocity head of the average velocity = inches of 32 F water column

v = absolute viscosity = lb per hr per ft = 242 poises

f. The loss of total pressure is then:

$h = qf$ (for shock losses)
$h = qf$ (for straight pipe a length equal to its diameter)

This latter for any length pipe is:

$$h = qf\,\frac{L}{D}$$

Obviously, the L/D makes the friction come out directly proportional to the length and L and D are in feet.

These coefficients vary, but it has been found (when reasonably below the speed of sound) that if we plot the test data for a given shape of object against a number called Reynolds number we can use that coefficient to get the friction with fluids other than those used on the test. This is true provided the physical object in question has the same proportionate shape and, within reason, is not too much larger or smaller than the object used to get the test data.

Again, consider the similarity between the friction on the inside of a pipe and the drag on an airplane wing. The two formulas look the same. The coefficients obtained on test are plotted against Reynolds number in both cases. In aircraft, to predict performance of a large wing in flight, you simply use a small model of the wing in a wind tunnel, operating it at the same Reynolds number. The test data will give a coefficient valid at that same Reynolds number in flight. In connection with pipes or ducts, we may use test data from a small pipe to predict performance in a larger one or one fluid on test to predict performance with another fluid.

Reynolds number should be estimated in the smallest section measurable. In an orifice, don't take it in the pipe but in the orifice because the velocity is higher there, not in the vena contracta because that is not readily measurable. For straight pipe use the pipe cross section, for an ell use the cross section having the smallest equivalent diameter. In general where you can measure the area of a section and where there is a choice of areas, pick the smallest. A short method for getting the Reynolds number in duct work is needed.

Equivalent Diameter —Quick Method

It has been found practical from tests to use data from round ducts to predict friction in ducts of rectangular and other shapes. In order to do this, ducts that are not round must be given an "equivalent diameter," D:

$$D = 1\tfrac{1}{3}\,\frac{\text{(sectional area, sq in.)}}{\text{(perimeter, in.)}} = \text{(feet)}$$

Upon inspection, it will be noted that this equation gives the diameter in feet if applied to a round pipe and if to a very narrow rectangle it works out to twice the narrow dimension in feet. For a square, it gives the dimension of one side in feet.

Knowing the above, and referring to Fig. 1, you will see how readily one could guess roughly at the

equivalent diameter of a duct. For a rectangle or circle, it is once, for an infinitely wide duct (parallel plates), it is twice the spacing of the plates. It is always between once and twice the smallest dimension.

Reynolds Number —Quick Method

Now for a quick method for getting the Reynolds number (R.N.) in a duct. For air, this boils down to:

$$(\text{R. N.}) = (\text{constant}) \times (\text{fpm}) \times D$$

The constant is given by the following tabulation:

Temperature:	0 F	70 F	140 F
Sea level	140	105	85
5,000 ft alt.	115	85	70
10,000 ft alt.	95	75	60

In using Reynolds number, it need only be known approximately. Note that for ordinary air at sea level, 100 times the fpm times the equivalent diameter in feet gives the approximate Reynolds number.

We now have Reynolds number available for use. Our purpose is to use it in picking friction coefficients from charts and from other data already built up for us by test methods. When a coefficient is plotted against R.N., we note that at some low R.N. there is a marked change in the coefficient. This is known as the transition point. Below this point we say the flow is laminar and above it we say it is turbulent. This point is not definite and on a given set-up the flow can shift from one form to the other through such slight causes as a slight jar on a pipe. On straight pipe, the shift usually occurs somewhere between a R.N. of 2000 and 4000. In the turbulent region at high R.N., the coefficient tends to approach a constant.

For shock losses the coefficient is fairly constant at the R.N. in the ranges we use with ducts. That is why you do not see an attempt to plot f' vs. R.N. for an elbow, for instance. We simply state a given resistance is, say, $\frac{1}{2}\ q$, knowing that in our R.N. range, f' is fairly constant at $\frac{1}{2}$. If we were to test such an obstruction in the laboratory it would be well, however, to plot the data against R.N.

On the other hand, the smooth flow found in ducts and pipes does not carry a constant coefficient in the R.N. range we use. It is generally above the transition point, but it has not yet reached the limit where it is fairly constant.

Estimating Loss in Straight Duct

Many hundreds of tests on round tubes have been correlated. An approximation of the results is shown in Fig. 2. The lines shown on the chart came from data gathered on tests made with many kinds of liquids and gases and using round tubes having diameters from 1/10 in. to 10 ft or more. It is surprising, when a test is run in the laboratory, how the test data fall on these curves just where it is expected.

Data on Fig. 2 may be used to figure the pressure loss in a duct or tube. This drop is given by:

$$h = qfL/D$$

We are now able to estimate q, we can guess the approximate R.N., and so can pick f from Fig. 2. The length in feet and the equivalent diameter in feet are obvious and in connection with pipe and duct dimensions we use the clear, exact inside dimensions.

Referring to Fig. 2, notice the top line—this is the roughest possible pipe, $f = 0.054$–. Suppose we had a corrugated pipe. Take D as the *clear* inside diameter, and to play safe assume the pipe to be the roughest possible. Then we'll let $f = 0.054$ and we won't be far wrong. I have had corrugated rubber hose plot points on test pretty close to $f = 0.054$–.

Our primary object here is metal duct work. Ordinary sheet metal reasonably smooth inside will fall on the lower line, a reasonably smooth pipe.

At this point, it is well to mention that the region we expect to be in is turbulent and that the test data on Fig. 2 represent a section of pipe in a long straight run. If we use it for estimating, we must use it for a *long* reasonably straight pipe or if for a short pipe we must arrange the flow so that it is straight and uniform entering and leaving the short pipe. This should be done anyway by properly designed ells and fittings. In certain cases these fittings should be vaned as will be discussed later. Suffice to say that the lower two lines in Fig. 2 are satisfactory for properly designed ducts.

In order that we may have a rough idea of the f value for ducts, let us assume a possible minimum and maximum found in practice—just rough guesses:

fpm	Diam. ft	R.N. = 100 (fpm) D	f from Fig. 2
200	$\frac{1}{4}$	$100 \times 200 \times \frac{1}{4} = 6660$	0.038
4000	10	$100 \times 4000 \times 10 = 4{,}000{,}000$	0.0095

We see therefore a possible range of f in duct practice of from 0.038 for small ducts and low velocities to 0.0095 for large ducts and high velocities. We see also that even with a small duct (4 in.) and a very low velocity we may still use the turbulent line.

It has been found in practice that the round pipe data in Fig. 2 may be used to predict pressure loss in rectangular pipes provided that for D we use the equivalent diameter and if for flat ducts we increase f by one-third in the laminar and one-fourth in the turbulent region. Accordingly, I have inserted in Fig. 2 a dotted cross line to be used for flat ducts.

Work was done on this by Atherton and was reported in the 1926 ASME Transactions, if you care to read up on it. I have personally conducted tests that check Fig. 2. I have even designed and tested a narrow duct 5 in. x ¼ in. in section, a 20 to 1 ratio that went into the wing of an airplane. The flow was in the laminar region and checked closely with the dotted cross laminar line in Fig. 2.

It is probably best to refer to Fig. 2 to get f, but if you care to memorize the approximate formula for the lower line for use above R.N. = 4000, it is:

$$f = \frac{1}{5\ (\text{R.N.})^{1/5}}$$

For the laminar line, which would rarely be used, for use below R.N. = 2000, it is:

$$f = 64/\text{R.N.}$$

It should be realized that all the test data and practice applies, using the lower line for smooth steel or similar surface. Smooth surface would mean flush joints in ducting.

The formula to memorize for any duct, any pipe, any fluid, is:

$$h = qfL/D$$

One of the points brought out in this article is the fact that shock losses, losses that are due to very

irregular and bumpy flow paths, changes of direction and so on, have a coefficient that reaches a constant at some low R.N. In the R.N. range in which we are interested, the coefficient generally remains constant. Losses due to fairly smooth flow—the loss in straight, smooth pipe—have a decreasing coefficient as the R.N. increases. They too, but at a much higher R.N., reach a constant coefficient. This difference is well illustrated in Fig. 2. The upper line made by the roughest possible pipe consists of shock losses. Notice that it reaches a constant 0.054 at a low R.N., 3000, and that it continues a constant in the range in which we operate. The lower line is the other extreme, the smooth flow. In the range we operate in, it has not yet reached a constant.

In designing duct work, plan to dump air into a straight piece of pipe with reasonably the same velocity across the section. A properly designed elbow or other obstruction will do this. A long straight pipe in the R.N. range we work in will have nearly the same velocity all across the entire section. If you therefore use properly designed obstructions you may use Fig. 2 for short pieces of duct. The lower smooth pipe line is then near enough to practical duct work for universal use.

In Fig. 2, there is shown in dotted lines certain other sizes of galvanized ducts that would be considered "rougher" pipe than the lower solid line. Considering, for instance a 3 in. tube and a velocity of 3000 fpm, we have an R.N. of 79,000 and $f = 0.0215$ on the lower solid line, which is more accurately $f = 0.023$ on the dash dot line, merely a 7 percent increase in f and hardly worth bothering about.

In the same way, consider a 1 in. duct and, say, 600 fpm. We have R.N. = 5250 and $f = 0.040$ on the solid line, which is more accurately $f = 0.043$ on the dotted line, a difference which is not important.

Both the above illustrations are the maximum roughness we might get in duct work that would suggest we ought to pick a point from the lower two lines. Obviously duct work should be specified with flush joints inside the smaller diameters. If this is done, it's quite obvious that the lower two heavy lines of Fig. 2 may be used for any practical duct problem.

2000 Ton System for New York Skyscraper

THE equivalent of "a 2000 ton iceberg melting every 24 hr" highlights the air conditioning system to be installed in New York City's newest skyscraper under construction at 100 Park Ave., according to Alfred L. Jaros, Jr., of Jaros, Baum & Bolles, consulting engineers.

The system will be part of the basic construction and will make the 36 story, $20 million structure an unrivalled, year 'round "oasis of atmospheric comfort" for tenants, according to Mr. Jaros, who declared that all areas of the building will be conditioned 12 months a year—by cooling and dehumidifying in summer, heating in winter, and ventilating at all seasons.

The system is featured by a unique zone arrangement for conditioning independently both inner and outer spaces of each floor. Designed by Jaros, Baum & Bolles, the zone layout will not only insure uniform effectiveness of the air conditioning but also will add to the flexibility of control. It will be possible to provide varying degrees of conditioning both to individual floors and to portions of those floors as they are affected by changing heat from the sun, conduction of heat through windows and walls, and wind.

The outer zones, or areas adjacent to the windows and extending inward for fifteen feet, will be provided with a system of peripheral conditioning. This will treat a quantity of outdoor air for ventilating purposes and distribute it through vertical riser pipes. Paralleling these pipes will be a set of supply and return pipes through which pumps and heat exchangers will circulate cold water in summer and hot water in winter to supply additional cooling, or heating, to these spaces as needed.

Under each window there will be an individual induction unit, concealed within the window sill, which will automatically mix, treat, and redistribute outdoor air and recirculated air from the room itself in sufficient quantity and proportion to maintain the atmospheric conditions desired by the occupants of that area. This operation will be controllable by an individual valve.

Each window will thus be, in effect, a complete module, with its own individual conditioning apparatus. No matter where a tenant may choose to locate dividing partitions for his office layout, each section will be provided with proper ventilation and air conditioning.

The interior zone system, for spaces more than 15 ft from the outside walls, will draw air from corridors or open spaces and mix it at the fan room with an equal amount of outdoor air. Properly filtered, cooled or warmed, the mixture will then be distributed by the main fan through ducts to strategically located outlets. A separate small fan room will serve every two or three floors up to the 20th, while the tower floors will contain a single primary fan room with local mixing units on each floor.

Steam from the mains of the New York Steam Co. will provide heating in winter, and, in summer, will be used to drive the centrifugal compressors to chill the water that will be pumped to all points where cooling is required.

Mechanical exhaust ventilation will be provided for washrooms and other spaces requiring such treatment. Air conditioning of spaces in the ground floor and basement will be provided by the tenants entirely independent of the main system and will make possible the operation of such spaces during hours when the office areas are unoccupied.

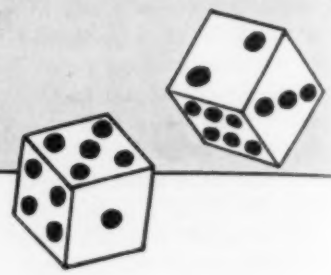
According to Mr. Jaros, the total amount of outdoor air supplied by the primary fans to the units will exceed 4 million cfh. The total amount of outdoor air supplied to the interior zones will exceed 7 million cfh. Including the recirculated air, approximately 41 million cfh of conditioned air will be circulated throughout the rentable spaces of the building.

The cooling towers will circulate 360,000 gph of water through the condensers of the refrigerating plant.

Mr. Jaros further pointed out that the maximum steam required for winter heating and ventilation will approximate 1500 boiler horsepower. Approximately two-thirds as much will be used in the hottest summer weather to operate the refrigerating plant.

Probability for Engineers

How to Use the Laws of Chance



A. B. Stickney, formerly engineer with Armour & Co., Chicago, now with The Rust Engineering Co., Pittsburgh, and a past national president of the American Society of Refrigerating Engineers, has made extensive use of probabilities in determining the amount of standby fuel needed when natural gas is bought on an interrruptible basis; he discussed this in our August '48 issue. This month, he explains fundamentals of probability

ENGINEERS are occasionally confronted with problems where it would be advantageous to evaluate probabilities. These come up when deviations from a mean value occur according to the laws of chance. This may be where numerous measurements of the same quantity are made, in which case deviations are errors. More frequently, in engineering work, deviations are differences of actual quantities from the average. Examples of this, applicable to his particular work, will occur to any engineer.

This type of problem is not of frequent enough occurrence to justify the average engineer in maintaining familiarity with the mathematical theory of probability. Rather, he wants a means of empirical application. The object of this article is to provide such an empirical approach, while maintaining the mathematical accuracy of the theoretical approach.

The hallmark of chance distribution is that the deviations follow the familiar probability curve pattern. Fig. 1 shows a set of deviations from normal temperature for the daily average temperature in Chicago for the year 1945. The height at each deviation shows the number of days when the actual mean temperature deviated by that amount from the "normal" mean temperature for that day.

Superimposed on it is a mathematically correct probability curve using the correct constants to fit it to these particular data. The agreement is good enough to permit the inference that the deviations follow the laws of chance. Any data which, when plotted, give a curve of this

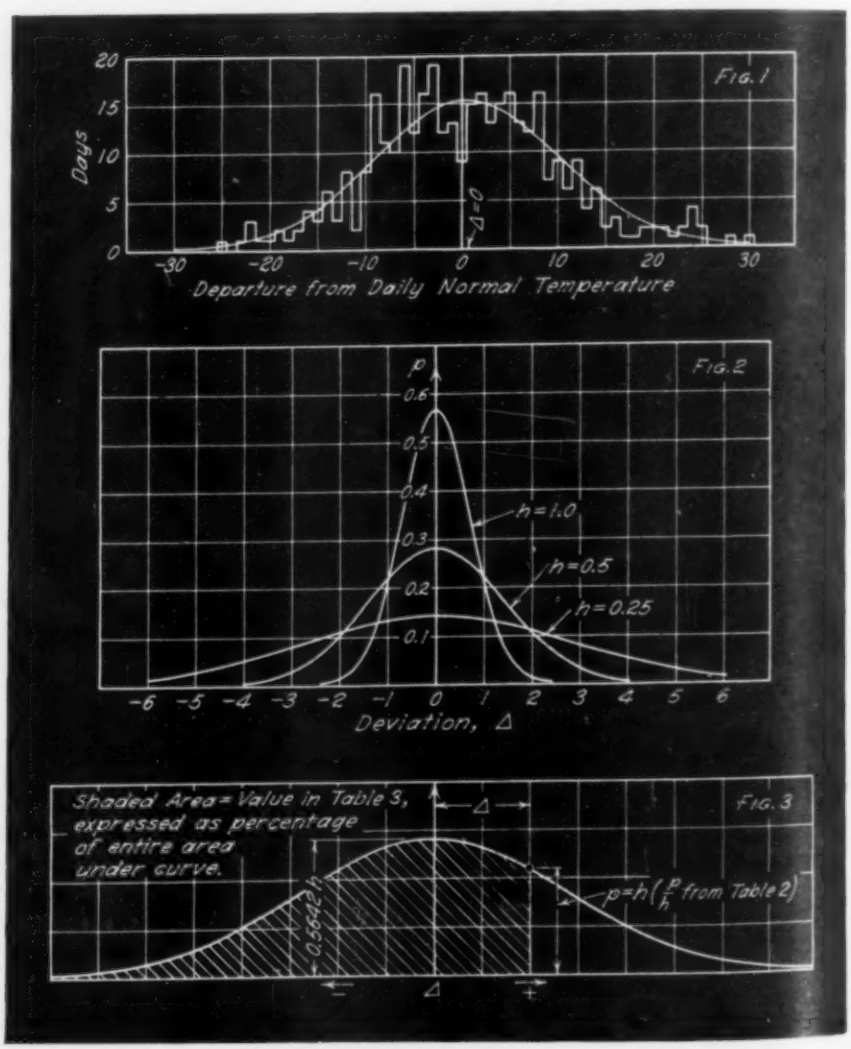


Fig. 1 (Top)—Departures from daily normal temperature in Chicago for the year 1945, and standard probability curve. Fig. 2 (Center)—Effect of the constant *h* on the shape of the probability curve. Fig. 3 (Bottom)—This sketch illustrates the use of Tables 2 and 3, as explained in text

Table 2

This is without regard to sign; that

Table 1 — Computation sheet for data of Fig. 1

Departure from normal temperature	No. days, n	$n \times$ departure	Departure − arith. mean Δ	$n \times \Delta$	$n \times \Delta^2$	Departure from normal temperature	No. days, n	$n \times$ departure	Departure − arith. mean Δ	$n \times \Delta$	$n \times \Delta^2$	$n_{+\Delta} + n_{-\Delta}$	Cumulative Total
+ 1	15	+ 15	+ 0.5	+ 7.5	3.75	0	9	0	− 0.5	− 4.5	2.25	24	24
2	16	32	1.5	24.0	36.00	− 1	13	− 13	− 1.5	− 19.5	29.25	29	53
3	13	39	2.5	32.5	81.25	− 2	12	− 24	− 2.5	− 30.0	75.00	25	78
4	15	60	3.5	52.5	183.75	− 3	19	− 57	− 3.5	− 66.5	231.75	34	112
5	16	80	4.5	72.0	324.00	− 4	16	− 64	− 4.5	− 72.0	324.00	32	144
+ 6	13	+ 78	+ 5.5	+ 71.5	393.25	− 5	12	− 60	− 5.5	− 66.0	363.00	25	169
7	12	84	6.5	78.0	507.00	− 6	19	−114	− 6.5	−123.5	802.75	31	200
8	16	128	7.5	120.0	900.00	− 7	10	− 70	− 7.5	− 75.0	562.50		
9	7	63	8.5	59.5	505.75	− 8	11	− 88	− 8.5	− 93.5	794.75		
10	10	100	9.5	95.0	902.50	− 9	16	−144	− 9.5	−152.0	1444.00		
+11	6	+ 66	+10.5	+ 63.0	661.50	−10	8	− 80	−10.5	− 84.0	882.00		
12	9	108	11.5	103.5	1190.25	−11	2	− 22	−11.5	− 23.0	264.50		
13	4	52	12.5	50.0	625.00	−12	8	− 96	−12.5	−100.0	1250.00		
14	6	84	13.5	81.0	1093.50	−13	3	− 39	−13.5	− 40.5	546.75		
15	2	30	14.5	29.0	420.50	−14	6	− 84	−14.5	− 87.0	1261.50		
+16	3	+ 48	+15.5	+ 46.5	720.75	−15	3	− 45	−15.5	− 46.5	720.75		
17	1	17	16.5	16.5	272.25	−16	4	− 64	−16.5	− 66.0	1089.00		
18	1	18	17.5	17.5	306.25	−17	2	− 34	−17.5	− 35.0	612.50		
19	2	38	18.5	37.0	684.50	−18	1	− 18	−18.5	− 18.5	342.25		
20	2	40	19.5	39.0	760.50	−19	2	− 38	−19.5	− 39.0	760.50		
+21	2	+ 42	+20.5	+ 41.0	840.50	−20	1	− 20	−20.5	− 20.5	420.25		
22	1	22	21.5	21.5	462.25	−21	1	− 21	−21.5	− 21.5	462.25		
23	2	46	22.5	45.0	1012.50	−22	3	− 66	−22.5	− 67.5	1518.75		
24	4	96	23.5	94.0	2209.00	−23	1	− 23	−23.5	− 23.5	552.25		
25	2	50	24.5	49.0	1200.50	−24	..	0	−24.5	− 0.0	0.00		
+26	..	+ 0	+25.5	+ 0.0	0.00	−25	1	−25	−25.5	− 25.5	650.25		
27	..	0	26.5	0.0	0.00	−26	..	0	−26.5	− 0.0	0.00		
28	1	28	27.5	27.5	756.25	−27	..	0	−27.5	− 0.0	0.00		
29	..	0	28.5	0.0	0.00	−28	..	0	−28.5	− 0.0	0.00		
30	1	30	29.5	29.5	870.25	−29	..	0	−29.5	− 0.0	0.00		
Totals	182	+1494		+1403.0	17923.50		183	−1309		−1400.5	15962.75		

Each Δ covers range from $(\Delta - \frac{1}{2})$ to $(\Delta + \frac{1}{2})$. Therefore 169 are the deviations between −6 and +6 and 31 are between ±6 and ±7. Of these $[(182 + 183)/2] - 169 = 13\frac{1}{2}$ are within the median Δ.

$$\text{Arithmetic mean} = \frac{+1494 - 1309}{182 + 183} = +0.507, \text{ say } +0.5$$

$$\text{Root mean square deviation} = \sqrt{\frac{17923.50 + 15962.75}{182 + 183}} = 9.635;\ h = \frac{0.7071}{9.635} = 0.0734$$

$$\text{Average deviation} = \frac{1403.0 + 1400.5}{182 + 183} = 7.681;\ h = \frac{0.5642}{7.681} = 0.0735$$

$$\text{Median deviation} = 6 + \frac{13\frac{1}{2}}{31} = 6.436;\ h = \frac{0.4769}{6.436} = 0.0741$$

$$\text{Probable error of arithmetic mean} = \pm \frac{0.4769}{0.0734\sqrt{365}} = \pm 0.341$$

$$\text{Probable error of } h = \frac{0.4769 \times 0.0734}{\sqrt{365}} = \pm 0.0018$$

general shape can have a probability curve fitted. Our problem is to obtain the constants which will best make the type curve fit the actual data.

The equation of the probability curve is

$$p = \frac{h}{\sqrt{\pi}} e^{-h^2\Delta^2}$$

where p = probability; Δ = deviation; h = constant which determines the shape of the curve.

For any value of h the area under the curve, from Δ equals minus infinity to Δ equals plus infinity, is unity. Fig. 2 shows the shape of the curve for several values of h. The greater the value of h, the higher the curve at the center, the steeper the sides, and the smaller the value of Δ at which p reaches small values. The problem is to find the value of h which will most nearly fit the particular data considered.

To do this, first find the *arithmetic mean* of all observations. For this, $\Delta = 0$. Next, find the difference of each observation from the arithmetic mean, square this difference, add the squares of all observations, divide by the number of observations, and take the square root of the result. This is the *root mean square deviation*. Then

$$h = \frac{0.7071}{\text{root mean square deviation}}$$

Table 1 illustrates this procedure for the data of Fig. 1.

Having h, Table 2 permits us to construct the probability curve. It shows, for various values of $h\Delta$, the corresponding value of p/h. For any value of Δ, multiply by h, read into the table to obtain p/h, and multiply this by h. This gives the probability p for that value of Δ, expressed as a decimal fraction. Fig. 3 illustrates this. The area under the curve derived this way, between $(\Delta - \frac{1}{2})$ and $(\Delta + \frac{1}{2})$ is the probability that a given deviation will be within $\pm\frac{1}{2}$ of Δ.

What we usually want is the probability that a given deviation will lie between Δ_1 and Δ_2. We want the percentage of all cases which will fall within these limits. We know that the total area under the probability curve is unity. We want the percentage of this area which will lie between Δ_1 and Δ_2.

Table 3 has been prepared to give this. For each value of $h\Delta$, it gives the percentage of the total area which lies between minus infinity and Δ. This is represented by the shaded area in Fig. 3. To find this percentage for Δ_1, multiply Δ_1 by h, read this value into the table, and obtain the percentage. Do the same for Δ_2. The difference between these percentages is the percentage of all observations which fall between Δ_1 and Δ_2.

Finally, how accurate is the formulation? We are dealing with probabilities. What are the probable errors in the derivation of our quantities?

The probable error is that value which, without regard to sign, is as likely to be exceeded as not. The probable error depends on the number of observations. The greater the number, the less the probable

Table 2

$$\frac{p}{h} = \frac{e^{-h^2\Delta^2}}{\sqrt{\pi}}$$

$h\Delta$	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
0.0	.56419	.56413	.56397	.56368	.56329	.56278	.56216	.56142	.56059	.55964
0.1	.55857	.55740	.55612	.55473	.55324	.55164	.54993	.54812	.54620	.54419
0.2	.54206	.53985	.53753	.53512	.53262	.53001	.52731	.52452	.52165	.51868
0.3	.51563	.51250	.50928	.50598	.50260	.49914	.49561	.49200	.48833	.48458
0.4	.48077	.47689	.47295	.46895	.46489	.46077	.45659	.45236	.44809	.44376
0.5	.43939	.43497	.43051	.42602	.42149	.41692	.41232	.40768	.40302	.39833
0.6	.39362	.38889	.38413	.37936	.37457	.36977	.36496	.36014	.35531	.35048
0.7	.34564	.34080	.33596	.33112	.32629	.32147	.31665	.31184	.30704	.30226
0.8	.29749	.29274	.28801	.28329	.27860	.27394	.26929	.26467	.26008	.25552
0.9	.25098	.24648	.24201	.23758	.23317	.22881	.22448	.22019	.21594	.21173
1.0	.20755	.20342	.19934	.19529	.19129	.18733	.18342	.17956	.17574	.17197
1.1	.16824	.16456	.16093	.15736	.15382	.15034	.14691	.14352	.14019	.13691
1.2	.13367	.13049	.12736	.12427	.12124	.11826	.11533	.11245	.10962	.10684
1.3	.10410	.10142	.09879	.09621	.09367	.09119	.08875	.08636	.08402	.08172
1.4	.07947	.07727	.07511	.07300	.07094	.06892	.06694	.06501	.06312	.06127
1.5	.05946	.05770	.05598	.05430	.05266	.05106	.04949	.04797	.04648	.04503
1.6	.04361	.04224	.04089	.03958	.03831	.03707	.03587	.03469	.03355	.03244
1.7	.03136	.03030	.02928	.02829	.02732	.02639	.02548	.02459	.02373	.02290
1.8	.02210	.02131	.02055	.01982	.01910	.01841	.01774	.01709	.01646	.01585
1.9	.01526	.01469	.01414	.01361	.01309	.01259	.01211	.01164	.01119	.01075

$h\Delta$	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
2.0	.01033	.00686	.00446	.00284	.00178	.00109	.00065	.00038	.00022	.00013

error. It is inversely proportional to the square root of the number of observations. The probable error of the arithmetic mean is

$$\pm \frac{0.4769}{h\sqrt{\text{no. of observations}}}$$

The probable error of h is

$$\pm \frac{0.4769\,h}{\sqrt{\text{no. of observations}}}$$

There is a 50-50 chance that the true values are this close to the computed values. The chances are better than 9 to 2 against their being more than twice this far from those computed. For the data of Fig. 1, Table 1 shows that the probable error of the arithmetic mean is ± 0.341 and of h is ± 0.0018.

For data comprising a large number of observations, these probable errors become small; for a small number of observations, say less than 100, they should be evaluated to determine their magnitude, to see whether they are significant in the use to be made of the formulation. In any event, the formulation is the most accurate which the data justify, and if greater accuracy is required, it can only be obtained by broadening the base on which the formulation lies (by accumulating a greater number of observations).

For large numbers of observations which follow the probability curve closely, two shortcuts are available. These are that

$$h = \frac{0.5642}{\text{average deviation}}$$

$$= \frac{0.4769}{\text{median deviation}}$$

where the *average deviation* is the average of all deviations without regard to sign, and the *median deviation* is one of such magnitude that there are as many actual deviations greater than it is as there are deviations smaller than it is. This is without regard to sign; that is, minus deviations and plus deviations of the same size are grouped together.

These short cuts should be avoided if precision is required unless the number of observations is very large, or the data follow the probability curve quite closely. They are, however, useful as checks on the correlation of the formulation and the data, and also for approximations where precision is not required.

For the data used in Fig. 1, Table 1 shows that

h by root mean square deviation $= 0.0734 \pm 0.0018$;

h by average deviation $= 0.0735$;

h by median deviation $= 0.0741$.

The close agreement shows that there is a good correlation between the data and the formulation. In this case the data are sufficient so that any one of the three would have yielded a close enough approximation for most engineering purposes.

Table 3—Percent probability that a deviation will not exceed ($h\Delta$)

$$\text{Percent probability} = \frac{100}{\sqrt{\pi}} \int_{-h\Delta}^{h\Delta} e^{-h^2\Delta^2} d(h\Delta)$$

$h\Delta$	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
−2.0	0.234	0.149	0.093	0.057	0.035	0.020	0.012	0.006	0.003	0.002
$h\Delta$	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
−1.9	0.360	0.345	0.331	0.317	0.304	0.291	0.278	0.267	0.255	0.244
−1.8	0.545	0.524	0.503	0.483	0.463	0.445	0.427	0.409	0.392	0.376
−1.7	0.811	0.780	0.750	0.721	0.694	0.667	0.641	0.616	0.592	0.569
−1.6	1.182	1.140	1.098	1.058	1.019	0.981	0.945	0.910	0.876	0.843
−1.5	1.694	1.636	1.580	1.524	1.471	1.419	1.369	1.320	1.273	1.227
−1.4	2.385	2.308	2.231	2.157	2.085	2.015	1.948	1.882	1.818	1.755
−1.3	3.300	3.197	3.097	2.999	2.905	2.812	2.722	2.635	2.550	2.467
−1.2	4.485	4.352	4.224	4.098	3.975	3.855	3.738	3.625	3.514	3.406
−1.1	5.990	5.824	5.661	5.502	5.346	5.194	5.045	4.900	4.758	4.620
−1.0	7.865	7.660	7.458	7.261	7.068	6.878	6.693	6.512	6.334	6.160
−0.9	10.155	9.906	9.662	9.422	9.187	8.956	8.729	8.507	8.289	8.075
−0.8	12.895	12.600	12.310	12.024	11.743	11.467	11.195	10.928	10.666	10.408
−0.7	16.110	15.767	15.429	15.096	14.766	14.442	14.123	13.809	13.500	13.196
−0.6	19.807	19.416	19.030	18.648	18.271	17.899	17.531	17.169	16.811	16.458
−0.5	23.975	23.538	23.105	22.677	22.253	21.834	21.419	21.009	20.604	20.203
−0.4	28.580	28.102	27.627	27.156	26.689	26.226	25.767	25.313	24.863	24.417
−0.3	33.569	33.055	32.544	32.036	31.532	31.031	30.534	30.040	29.550	29.063
−0.2	38.865	38.324	37.785	37.249	36.715	36.184	35.655	35.129	34.606	34.086
−0.1	44.377	43.819	43.262	42.707	42.153	41.600	41.050	40.501	39.953	39.408
−0.0	50.000	49.436	48.872	48.308	47.744	47.181	46.619	46.057	45.496	44.936
+0.0	50.000	50.564	51.128	51.692	52.256	52.819	53.381	53.943	54.504	55.064
+0.1	55.623	56.181	56.738	57.293	57.847	58.400	58.950	59.499	60.047	60.592
+0.2	61.135	61.676	62.215	62.751	63.285	63.816	64.345	64.871	65.394	65.914
+0.3	66.431	66.945	67.456	67.964	68.468	68.969	69.466	69.960	70.450	70.937
+0.4	71.420	71.898	72.373	72.844	73.311	73.774	74.233	74.687	75.137	75.583
+0.5	76.025	76.462	76.895	77.323	77.747	78.166	78.581	78.991	79.396	79.797
+0.6	80.193	80.584	80.970	81.352	81.729	82.101	82.469	82.831	83.189	83.542
+0.7	83.890	84.233	84.571	84.905	85.234	85.558	85.877	86.191	86.500	86.805
+0.8	87.105	87.400	87.690	87.976	88.257	88.533	88.805	89.072	89.334	89.592
+0.9	89.845	90.094	90.338	90.578	90.813	91.044	91.271	91.493	91.711	91.925
+1.0	92.135	92.340	92.542	92.739	92.932	93.122	93.307	93.488	93.666	93.840
+1.1	94.010	94.176	94.339	94.498	94.654	94.806	94.955	95.100	95.242	95.380
+1.2	95.515	95.648	95.776	95.902	96.025	96.145	96.262	96.375	96.486	96.595
+1.3	96.700	96.803	96.903	97.001	97.095	97.188	97.278	97.365	97.450	97.533
+1.4	97.615	97.692	97.769	97.843	97.915	97.985	98.052	98.118	98.182	98.245
+1.5	98.306	98.364	98.420	98.476	98.529	98.581	98.631	98.680	98.727	98.773
+1.6	98.818	98.860	98.902	98.942	98.981	99.019	99.055	99.090	99.124	99.157
+1.7	99.189	99.220	99.250	99.279	99.306	99.333	99.359	99.384	99.408	99.432
+1.8	99.455	99.476	99.497	99.517	99.537	99.555	99.573	99.591	99.608	99.624
+1.9	99.640	99.655	99.669	99.683	99.696	99.709	99.722	99.733	99.745	99.756
$h\Delta$	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
+2.0	99.766	99.851	99.907	99.943	99.965	99.980	99.988	99.994	99.997	99.998

DATA SHEET

How to Choose Economical Type of Roof Construction

HARRY J. SCHARRES, mechanical engineering dept., Graham, Anderson, Probst and White, architects, Chicago, presents here tables that show the heat transfer coefficients for eight commonly used roof decks with seven different kinds of insulation. These tables have been found of value in selecting economical roof construction when considering heat loss or gain

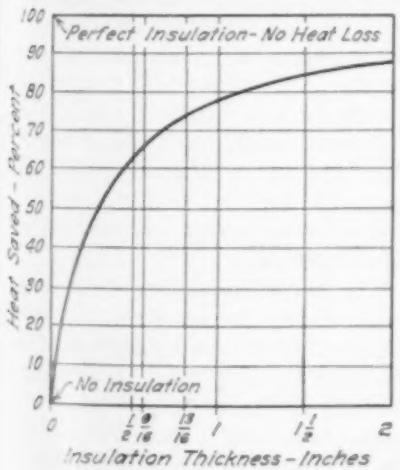
MOST ARCHITECTS will agree that, with the prevailing cost of buildings, the problem of economical design is one of increasing complexity. Since each project presents new or different problems, no governing rules for achieving economy can be set that would apply to all types of construction.

The procedure to be followed is one of correlation between the architects and the structural and mechanical engineers in order that the most economical overall design can be decided upon. No one phase of building design can be executed separately and in disregard of the other problems.

Tables Indicate Possible Savings

The heating and air conditioning designer plays an increasingly important role in the overall building design, and he is frequently called upon to make recommendations to the architect on how the initial and operating costs may be reduced. The usual procedure in such a case

Typical variation of heat saved with insulation thicknesses. Note: All data and symbols used in connection with the accompanying tables were taken from the ASHVE Heating, Ventilating, Air Conditioning Guide 1947, chapter 6



KAYLO TILE ROOF DECK
Surface Film C=6.00
Built-up Roof C=3.53
Insulation k=as Shown Below
Kaylo Roof Tile k=0.62
Surface Film C=1.65

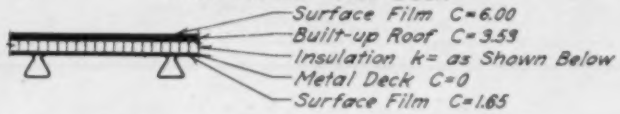
Coefficients of Heat Transmission (U) in Btu per (hr)(sq ft)(°F diff.)—Air to Air

Insulation Thickness, Inches	2⅜" Slab No Insulation, U=0.19						
	Foamglas k=0.40	Fiberglas k=0.28	Insulating Board k=0.29	Celotex k=0.33	Cork k=0.25	Cork k=0.30	Cork k=0.34
½	—	—	0.14	0.15	—	—	—
9/16	—	0.14	—	—	—	—	—
13/16	—	0.12	—	—	—	—	—
1	—	0.11	0.11	0.12	0.11	0.12	0.12
1½	—	0.09	0.10	0.10	0.09	0.10	0.10
2	0.10	0.08	0.08	0.09	0.07	0.08	0.09

PRECAST CEMENT TILE ROOF DECK
Surface Film C=6.00
Built-up Roof C=3.53
Insulation k as Shown Below
Precast Cement Tile k=12.00
Surface Film C=1.65

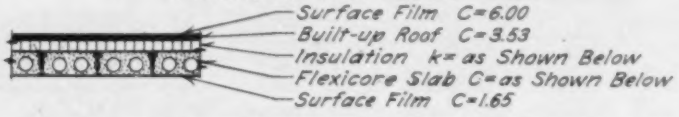
Coefficients of Heat Transmission (U) in Btu per (hr)(sq ft)(°F diff.)—Air to Air

Insulation Thickness Inches	1⅝" Slab No Insulation, U=0.84						
	Foamglas k=0.40	Fiberglas k=0.28	Insulating Board k=0.29	Celotex k=0.33	Cork k=0.25	Cork k=0.30	Cork k=0.34
½	—	—	0.34	0.37	—	—	—
9/16	—	0.31	—	—	—	—	—
13/16	—	0.24	—	—	—	—	—
1	—	0.21	0.22	0.24	0.19	0.22	0.24
1½	—	0.16	0.16	0.17	0.14	0.16	0.18
2	0.16	0.12	0.12	0.14	0.11	0.13	0.14



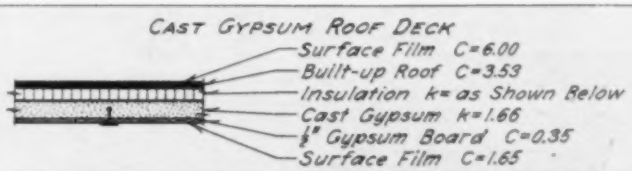
Coefficients of Heat Transmission (U) in Btu per (hr)(sq ft)(°F diff)—Air to Air

Insulation Thickness Inches	Any Type Metal Deck No Insulation, U=0.94						
	Foamglas	Fiberglas	Insulating Board	Celotex	Cork		
	k=0.40	k=0.28	k=0.29	k=0.33	k=0.25	k=0.30	k=0.34
½	—	—	0.36	0.39	—	—	—
⅝	—	0.33	—	—	—	—	—
¾	—	0.25	—	—	—	—	—
1	—	0.22	0.22	0.24	0.20	0.23	0.25
1½	—	0.16	0.16	0.18	0.14	0.17	0.18
2	0.17	0.12	0.13	0.14	0.11	0.13	0.14



Coefficients of Heat Transmission (U) in Btu per (hr)(sq ft)(°F diff.)—Air to Air

Insulation Thickness	6"Slab-Sand & Stone Aggregates, C=1.25 No Insulation, U=0.54							6"Slab-Lightweight Aggregates, C=0.89 No Insulation, U=0.46						
	Foamglas	Fiberglas	Insulating Board	Celotex	Cork			Foamglas	Fiberglas	Insulating Board	Celotex	Cork		
Inches	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34
½	—	—	0.28	0.30	—	—	—	—	—	0.26	0.27	—	—	—
⅝	—	0.26	—	—	—	—	—	—	0.24	—	—	—	—	—
¾	—	0.21	—	—	—	—	—	—	0.20	—	—	—	—	—
1	—	0.18	0.19	0.20	0.17	0.19	0.21	—	0.17	0.18	0.19	0.16	0.18	0.20
1½	—	0.14	0.14	0.16	0.13	0.15	0.16	—	0.13	0.14	0.15	0.12	0.14	0.15
2	0.15	0.11	0.11	0.13	0.10	0.12	0.13	0.14	0.11	0.11	0.12	0.10	0.11	0.12



Coefficients of Heat Transmission (U) in Btu per (hr)(sq ft)(°F diff.)—Air to Air

Insulation Thickness	2½"Slab Including ½" Gypsum Board No Insulation, U=0.38							3½"Slab Including ½" Gypsum Board No Insulation, U=0.31						
	Foamglas	Fiberglas	Insulating Board	Celotex	Cork			Foamglas	Fiberglas	Insulating Board	Celotex	Cork		
Inches	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34
½	—	—	0.23	0.24	—	—	—	—	—	0.20	0.21	—	—	—
⅝	—	0.22	—	—	—	—	—	—	0.19	—	—	—	—	—
¾	—	0.18	—	—	—	—	—	—	0.16	—	—	—	—	—
1	—	0.16	0.16	0.18	0.15	0.17	0.18	—	0.15	0.15	0.16	0.14	0.15	0.16
1½	—	0.13	0.13	0.14	0.12	0.13	0.14	—	0.12	0.12	0.13	0.11	0.12	0.13
2	0.13	0.10	0.10	0.12	0.09	0.11	0.12	0.12	0.10	0.10	0.11	0.09	0.10	0.11

would be to spend hours of calculating to produce results which can be understood by the architect and the client.

It has been found desirable to prepare a set of tables that indicate clearly to the architectural and structural designers (as well as to the client) the savings that can be gained in the heating and air conditioning phase by choosing certain methods of construction. Such tables dealing with roof construction are presented here; they cover eight of the most used types of roof decks and seven different kinds of insulation, in thicknesses up to 2 in. Each table is complete in itself, and gives the heat transfer coefficient *U* (Btu per hr per sq ft per deg temperature difference from air to air) for the kind of insulation and the thickness selected. *k* is thermal conductivity, Btu per hr per sq ft per F per in., and *C* is thermal conductance, Btu per hr per sq ft per F.

All data used were taken from the ASHVE's Heating, Ventilating, Air Conditioning Guide 1947, chapter 6. In all eight types of roof construction, the same built-up roof and the same inside and outside surface film conductances were used.

Although a number of additional types of roof construction and insulations could have been included, it was found to be unnecessary since most other possibilities would have been a duplication of the tables presented.

Economical Limits of Insulation Thickness

The graph showing the variation of heat saved with insulation thickness is particularly valuable in helping the client or the engineer to arrive at the economical limits of thickness of insulation. It can be seen from this graph that ½ in. of insulation would generally be the lower limit and to recommend more than 1 in. thickness would be doubtful from an economic standpoint, unless special considerations are involved.

The tables presented here were prepared by the author in the mechanical engineering department of the architectural firm of Graham, Anderson, Probst and White, Chicago, and under the direction of C. A. Frazier and R. H. Stevenson.

Concrete Roof Deck — Sand & Gravel Aggregate

Coefficients of Heat Transmission (U) in Btu per (hr)(sq ft)(°F diff.)—Air to Air

Insulation Thickness	2" Slab No Insulation, U=0.82							4" Slab No Insulation, U=0.72							6" Slab No Insulation, U=0.65						
	Foamglas	Fiberglas	Insulating Board	Celotex	Cork			Foamglas	Fiberglas	Insulating Board	Celotex	Cork			Foamglas	Fiberglas	Insulating Board	Celotex	Cork		
Inches	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34
1/8	—	—	0.34	0.36	—	—	—	—	—	0.32	0.34	—	—	—	—	—	0.31	0.33	—	—	—
9/16	—	0.31	—	—	—	—	—	—	0.30	—	—	—	—	—	—	0.28	—	—	—	—	—
13/16	—	0.24	—	—	—	—	—	—	0.23	—	—	—	—	—	—	0.22	—	—	—	—	—
1	—	0.21	0.21	0.24	0.19	0.22	0.24	—	0.20	0.21	0.23	0.19	0.21	0.23	—	0.20	0.20	0.23	0.18	0.21	0.22
1 1/2	—	0.15	0.16	0.17	0.14	0.16	0.18	—	0.15	0.15	0.17	0.14	0.16	0.17	—	0.15	0.15	0.17	0.13	0.15	0.17
2	0.16	0.12	0.12	0.14	0.11	0.13	0.14	0.16	0.12	0.12	0.13	0.11	0.12	0.14	0.15	0.12	0.12	0.13	0.10	0.12	0.13

Concrete Roof Deck—Lightweight Aggregate

Coefficients of Heat Transmission (U) in Btu per (hr)(sq ft)(°F diff.)—Air to Air

Insulation Thickness	2" Slab No Insulation, U=0.54							4" Slab No Insulation, U=0.38							6" Slab No Insulation, U=0.29						
	Foamglas	Fiberglas	Insulating Board	Celotex	Cork			Foamglas	Fiberglas	Insulating Board	Celotex	Cork			Foamglas	Fiberglas	Insulating Board	Celotex	Cork		
Inches	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34
1/8	—	—	0.28	0.30	—	—	—	—	—	0.23	0.24	—	—	—	—	—	0.19	0.20	—	—	—
9/16	—	0.26	—	—	—	—	—	—	0.21	—	—	—	—	—	—	0.18	—	—	—	—	—
13/16	—	0.21	—	—	—	—	—	—	0.18	—	—	—	—	—	—	0.16	—	—	—	—	—
1	—	0.18	0.19	0.20	0.17	0.19	0.21	—	0.16	0.16	0.18	0.15	0.17	0.18	—	0.14	0.14	0.15	0.13	0.15	0.16
1 1/2	—	0.14	0.14	0.16	0.13	0.15	0.16	—	0.12	0.13	0.14	0.12	0.13	0.14	—	0.11	0.12	0.12	0.11	0.12	0.13
2	0.15	0.11	0.11	0.13	0.10	0.12	0.13	0.13	0.10	0.10	0.11	0.09	0.11	0.12	0.12	0.09	0.10	0.10	0.09	0.10	0.11

Wood Roof Deck

Coefficients of Heat Transmission (U) in Btu per (hr)(sq ft)(°F diff.)—Air to Air

Insulation Thickness	1" Boards, 25/32" Actual No Insulation, U=0.49							2" Plank, 1 3/8" Actual No Insulation, U=0.32							3" Plank, 2 5/8" Actual No Insulation, U=0.23						
	Foamglas	Fiberglas	Insulating Board	Celotex	Cork			Foamglas	Fiberglas	Insulating Board	Celotex	Cork			Foamglas	Fiberglas	Insulating Board	Celotex	Cork		
Inches	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34	k=.40	k=.28	k=.29	k=.33	k=.25	k=.30	k=.34
1/8	—	—	0.26	0.28	—	—	—	—	—	0.21	0.22	—	—	—	—	—	0.16	0.17	—	—	—
9/16	—	0.25	—	—	—	—	—	—	0.20	—	—	—	—	—	—	0.16	—	—	—	—	—
13/16	—	0.20	—	—	—	—	—	—	0.16	—	—	—	—	—	—	0.14	—	—	—	—	—
1	—	0.18	0.18	0.20	0.17	0.19	0.20	—	0.15	0.15	0.16	0.14	0.16	0.17	—	0.13	0.13	0.14	0.12	0.13	0.14
1 1/2	—	0.14	0.14	0.15	0.12	0.14	0.16	—	0.12	0.12	0.13	0.11	0.12	0.13	—	0.10	0.11	0.11	0.10	0.11	0.11
2	0.14	0.11	0.11	0.12	0.10	0.12	0.13	0.12	0.10	0.10	0.11	0.09	0.10	0.11	0.11	0.09	0.09	0.10	0.08	0.09	0.10

Big Fan for Air Conditioned Store

Hoist 16 Ft High Blower to Sixth Floor

A 16 FT HIGH double width, double intake centrifugal fan was hoisted to the sixth floor of the new Bond clothing store, which is under construction in Chicago, last month. This blower will be driven by a 100 hp motor; its outlet dimensions are 107 by 74 in., and the inlets are 81 in. in diameter. The hoisting operation was supervised by Robert Irsay, of Acord Ventilating Co., the ventilating and sheet metal subcontractors for the project.

The fan—of welded construction—handles 115,000 cfm and weighs 8300 lb. Because of its construction, no special supports were necessary to protect it during the hoisting.

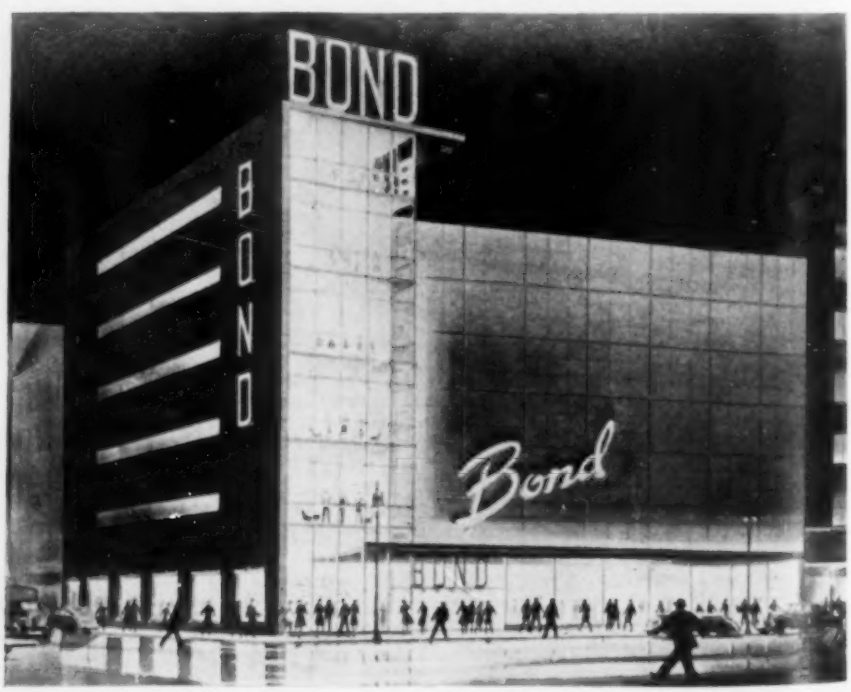
One of the outstanding air conditioning projects under way at the time, this new store occupies 100 ft on Jackson Blvd. and 125 ft on State St., a prominent "loop" corner. The building is of modern design, of concrete and steel fireproof construction; the exterior is glass brick and granite. A feature will be a continuous window on the State St. exposure.

The main supply conditioner will be in the sixth floor machinery room, will distribute air to the different parts of the building by a system of zone controlled ducts, with facilities for heating and cooling the entire building.

The building was designed by Friedman, Alschuler & Sincere, architects with Morris Lapidus, specialist on exteriors and interiors of buildings, associate. Robert E. Hattis is the consulting engineer, the general contractor is Sumner S. Sollitt & Co., and Thomas J. Douglass & Co. is the air conditioning, heating, and ventilating contractor, with the Acord company being awarded the ventilating and sheet metal subcontract.

(Photos courtesy New York Blower Co. Photo at left by Chicago Tribune)

Left—A big fan, with an outlet 107 by 74 in., is hoisted to the sixth floor of the new Bond store under construction in Chicago. Below—The air conditioned store —at State and Jackson—is scheduled for opening some time before Easter



SAM LEWIS' PAGE

Informal comment on heating, piping, and air conditioning matters is given in this regular feature by Samuel R. Lewis, consulting mechanical engineer, member HPAC's board of consulting and contributing editors

Tracking Down Troubles from Mysterious Moisture

MYSTERIOUS moisture in new buildings has been worrying several of my friends. Requests for diagnoses have come recently from widely separated cities.

In a northern city, a one story building has no basement and rests on a concrete slab. The heat transmission is from warm water to air via pipe coils imbedded in the floor.

In one rather large room, having several windows, there are a gas burning cast iron heating system boiler, a gas burning domestic hot water boiler, an electric water circulating pump, and the usual hot water storage tank, expansion tank, etc. The masonry chimney, to which the breechings are connected, is in the interior wall opposite the windows. When I visited the building I found the wall of the chimney saturated with water. The insulation on the two heaters and that on the tanks was as wet as an unwrung sponge. All metal work was heavily rusted. The plant had been in operation for about six months.

Testimony of the occupants disclosed that since the boiler room was a thoroughfare passage and since a window, when open, caused uncomfortable chilling, the window was kept closed.

A large volume of air is necessary to dilute the very heavily saturated, complex mixture of products of combustion from gas burners. For this reason the breechings from such heaters almost universally have air bypass inlets between the heaters and the chimneys. Air is encouraged to flow along with and to dilute the warm gases to the end that the mixture shall have such a high dew point temperature that moisture troubles in the chimneys will be avoided. In this plant, barely enough air could be obtained to support combustion.

The trouble was corrected by partitioning-off the thoroughfare and providing a large air intake duct to the boiler room directly from outside and near the ceiling.

Another case was in a southern city in a similar modern and very new single story basementless building. The heater is in an interior closet-like room, and it also burns gas.

The windows throughout the building were frosted in cold weather and dripped with condensate even in mild weather. The window stools or ledges looked as though they had just been washed and that wiping off the water had been neglected. The hardwood floors gave notice of imminent expansion and bowing up due to moisture. Many experiments had been carried out as to changes in temperature controls, and the builder had been suspected of improper drains under the footings, or of leaky water pipes concealed in the walls.

Stockpiles of steel in a warehouse will assume the outside air temperature during a short cool spell even in summer. There may then follow a quick change to warm, damp weather—during which the steel temperature is below the dew point temperature of the air. The steel then gets wet, rusting follows, and merchantability of the steel is reduced.

Outdoor Air Blocked Off

In the case of the interior of the above-mentioned building, variations in the outdoor temperature and relative humidity could not be blamed, because there were no variations observable in the steady accumulation of excess moisture and accompanying deposits of dew.

Use Halogen Leak Detector in Testing 13 Miles of Tubing

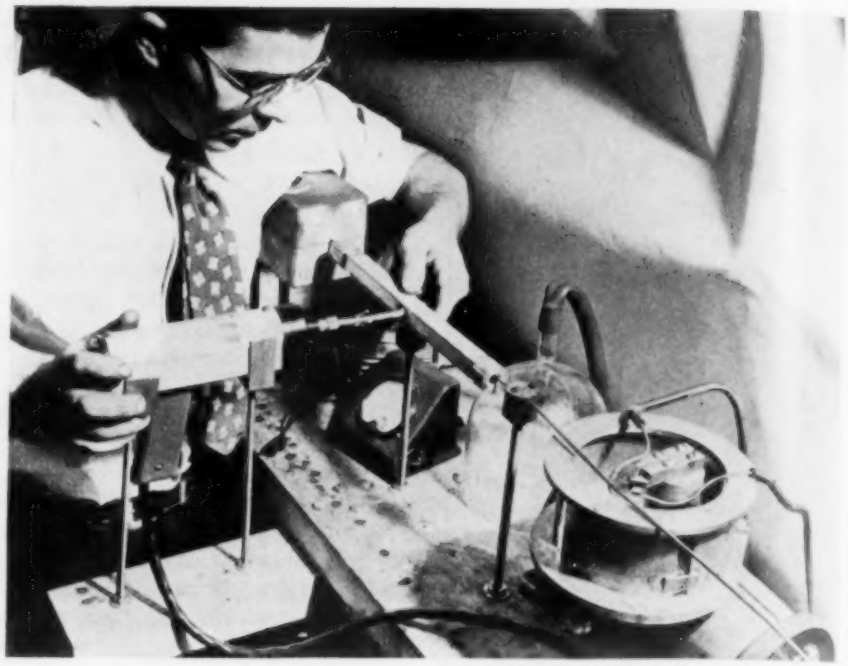
A HALOGEN type leak detector was employed recently to test for leaks in 13 miles of copper tubing being used for a special project at the General Electric Co.'s research laboratory.

According to John S. Hickey, a research laboratory engineer, this application of the instrument was necessitated by the laboratory's requirement for a large quantity of copper tubing completely free from leaks. The tubing, intended for use as a water cooled conductor to operate in a vacuum chamber, has a ⅛ in. outside diameter with a 0.015 in. wall. It was obtained in 100 ft lengths and induction brazed into mile-long lengths for use.

According to Mr. Hickey, a mixture of eight parts of nitrogen to one part of "Freon" was bled into the tubing, while the far end remained open, until samples of the escaping gas were found to contain "Freon," as shown by the leak detector. A valve on the end of the tubing was then closed and the pressure built up to 600 psi. The preliminary bleeding of the air was necessary, according to Mr. Hickey, to prevent the tubing at the far end of the coil from retaining only air.

Motor Driven Winding Drum

Except for about 5 ft at the ends of the coils, which were tested by hand, the actual leak test was conducted by running the tubing through a special pickup at about 1 fps, using a motor driven winding drum and a simple friction brake on the supply drum of the set-up.



Except for about 5 ft at the ends of the coils, which were tested by hand, the actual leak test was conducted by running the tubing through a special pick-up, using a motor driven winding drum and a friction brake on the supply drum

In the first test (on 13 miles of tubing having approximately 675 joints) seven leaky joints, two porous sections of tubing, and three folds were found. The leaky joints were poor braze jobs; the porous sections were 100 ft coils that had many leaks per ft over the entire length. The folds apparently were ingot flaws that closed up on drawing the tubing and were not visible to a casual inspection with the naked eye, according to Mr. Hickey.

Each reel was given a final check by putting it, fully charged with gas, into a cardboard box for a week. The box air was then tested for "Freon." One reel was found to have a leak. It was re-run on the test bench and a leaky joint found. This joint may have cracked when it was wound on the reel after the first test.

Subsequent experience with the tubing has shown that the leak detector found all the leaks that had existed.

SAM LEWIS' PAGE—

[*Concluded from preceding page*]

It was found, however, that an opening in the floor of the heater room, approximately equal in area to the chimney, had been provided for air to enter the otherwise air-tight room. The air was supposed to come from outdoors through leakage into the very shallow space between the bottoms of the joists and the ground. There was a metal grille over the opening in the floor that restricted the air flow, and mops, brooms, and even empty buckets stopped the air intake for long periods. Thus, while the gas burners received a starvation ration of air to support combustion, there was no air left for diluting the products of combustion and this nearly-saturated mixture condensed inside the chimney and dripped back to supply excess water through the walls.

I could say much more concerning the present trend (especially in small residences) toward placing good sized boilers or furnaces in little interior closets without considering the vital necessity for unobstructable, copious outside air supply.

No one should need to say more concerning the necessity for water-tight chimneys that have the absolute minimum of surface exposed to outside temperature. A chimney that is warmer than the dew point temperature will not condense vapor and thus will not be a wet chimney.

QUESTION OF THE MONTH

YOU ARE invited to submit a heating, piping, or air conditioning question for publication here. You are also invited to submit answers to previously-published questions from others. Those published are paid for at regular rates. Address the Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2, Ill.

Readers Comment on Printer's Problem of Air Conditioning

In the January issue, a printing company executive asked for information on the air conditioning of lithographing plants. While his question was such that a complete answer is hardly possible, the following comments are interesting and, we trust, helpful to him and to other printers facing the same situation

As THE QUESTION of the Month in the January HPAC, we published a letter from "D.S.", an executive of a printing company, who posed an interesting—perhaps a disturbing—problem concerning the air conditioning of lithographing establishments.

In brief, "D.S." said that his company had postponed, at least temporarily, plans for air conditioning. "We are looking for an air conditioning system that has a moderate amount of air change, humidity control, and temperature reduction," he said. "I recently came back from a convention of the National Association of Photo Lithographers. We had a speaker on air conditioning. The findings that I came away with from all this was that if you had an unlimited amount of money to spend for a super deluxe system with tolerances unnecessarily close, it could be had. But for the ordinary printer, such as ourselves, who has perhaps 10 to 25 bad days a year at which time the humidity actually injures our production, they don't have any intermediate systems to offer—only the finest.

"We believe that a humidity and temperature control with a reduction of 10 to 20 deg is sufficient. Lithographic prints can be made very nicely in a humidity of 60 to 65 percent, in our opinion. You do not have to spend extra thousands of dollars for 42 percent. Of course, this is our reaction to the information available to us," he concluded.

A number of comments on the subject have been received from readers, some of which are published here.

EACH PLANT MUST BE ANALYZED

EVERYTHING that "D.S." says is substantially true; he seems to be a man of wide experience. However, I don't think he has gone into the economics of air conditioning for printing plants far enough to establish some of the statements he makes.

Each plant has to be analyzed to determine how far it can economically go with air conditioning equipment. Certain plants, where the production is under definite control, insist on complete all year air conditioning control. In the larger plants, the flow of materials through various departments must be maintained in order to assure production in all of the departments. Production in one department cannot be stopped without affecting production in other departments.

In such plants described above, it has proved economical to provide all year air conditioning. This is substantiated by the fact that practically all plants that have all year air conditioning would under no condition expand other operations without expanding their air conditioning facilities.

In the smaller plants—and frequently in the larger plants—there seems to be a reasonable argument to provide air conditioning to take

care of winter, spring, and fall and not to be concerned too much with some delays during the summer. I don't think these people have checked thoroughly to see what it costs them to upset their production schedules.

If the operation is such that the jobs can be juggled around, they can manage quite well. If not, they have not taken the trouble to study their costs thoroughly. There is a fine dividing line and all I can say is this: If they can only see their way clear to air conditioning the plant for 90 percent of weather control, they have made a very substantial step in the right direction.

With regard to printing with air conditioning, please note that printing has been done for hundreds of years without temperature and humidity control and such printers have done a fairly good job. But once printers become accustomed to air conditioning control, and see the benefits of elimination of waste and delays, they will never want to go back to a non-air conditioned operation. The plant equipment and labor costs are too high to permit waste in manufacturing.

With regard to the desired humidity, you can run between 65 percent down to as low as 45 percent. However, it is important that once the humidity suitable for the operations has been selected, this humidity should be maintained.

I would like to add the following. All that "D.S." has to do is to listen to the complaints from the operators in a plant that is equipped with air conditioning, when the air conditioning system goes out of service. It just seems they cannot operate at all. It is true they manage to put the work through; however, they do it with considerable delay and fussing. They have forgotten how to take care of all of these variables that they once experienced.—B.O.

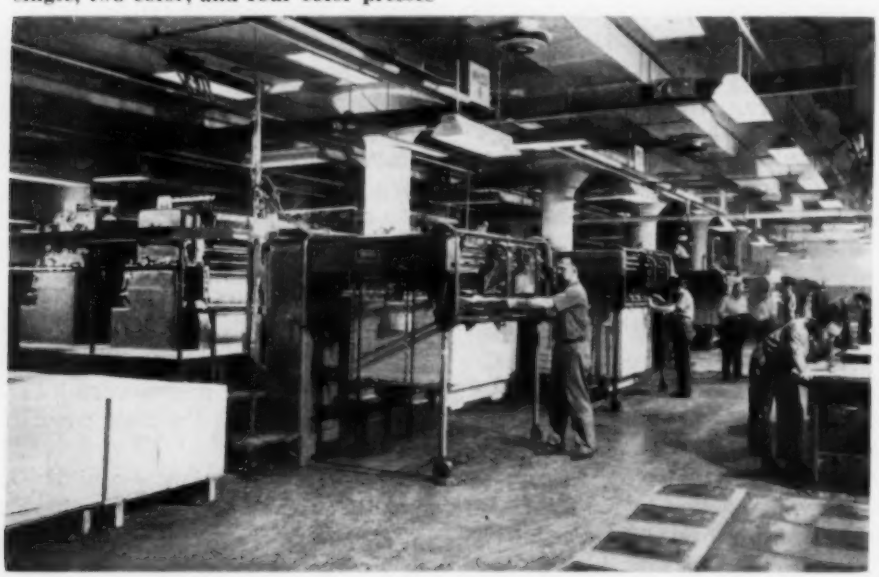
INCOMPLETE STATEMENT OF PROBLEM

THERE IS question as to how far we should go in answering the question from "D.S.," as a proper answer would require an article. The high spots are as follows:

"D.S." has not stated why he desires humidity control—whether his problem is one of register, cockling, difficulty with paper feeders, or what have you. Paper manufacturers, for the most part, ship paper which is relatively dry and, in order to keep it under control, the humidity must be kept reasonably low. In fact, for certain types of offset, it is desirable that the room be conditioned to a relative humidity which represents an equilibrium lower than the actual moisture content of the paper.

Printers should be advised that air conditioning a press room to some constant relative humidity will not completely solve the problem of register unless the paper is preconditioned.

Air conditioning promotes efficiency of operation at a big lithographing plant, where some 250 to 300 million color impressions are produced each year on single, two color, and four color presses



Years ago, printers used to talk about maintaining 65 percent relative humidity. They arrived at this magic number because it was the lowest relative humidity they could hope to hold constant in the summer without burning up their employees. If "D.S." would like relief on just a few days, he can get it without refrigeration if he is willing to use heat. Since the sensible heat is picked up by the air stream, the higher the relative humidity the more air is required, so that in selecting design conditions it is necessary to balance the cost of refrigeration against the cost of air distribution.

For a press room conditioning job in 1930 (described in the December 1931 HPAC), I decided that the compromise between the desirable low relative humidity and cost was about 45 percent. I believe that the latest recommendations are approaching this point.

I wish "D.S." were correct in the statement, "They [the air conditioning industry] don't have any intermediate system to offer, only the finest." If he will request competitive bids based on the opinions he expressed, and not on detailed plans, I am sure that he will find at least one that is bad enough to meet his requirements.—C.S.L.

50 to 55 PERCENT RELATIVE CORRECT RANGE

THE LETTER from "D.S." in the January issue about air conditioning for lithographing plants prompts me to submit the following observations. We recently made a study of this same problem for a local lithographer doing multicolor work requiring close register.

Paper, being a hygroscopic substance, may either gain or lose moisture to the atmosphere unless a condition of equilibrium exists between the paper and the atmosphere. It is therefore common practice to hang paper in paper conditioners, and bring the paper into equilibrium with the press room air by blowing press room air over the paper. This operation may either remove moisture from the paper or add moisture to it. The basic question then becomes one of conditioning the paper to that point where the least dimensional change will take place when additional moisture is either added or removed during

subsequent printing operations.

Weber and Geib of the U.S. Bureau of Standards have shown that at 4.6 percent moisture content, paper will stretch about twice as much per 0.1 percent increase in moisture content than paper at 5.8 percent, with the lowest rate of dimensional change occurring at about 5.5 percent. Since 5.5 percent moisture content paper is in equilibrium with air of about 51 percent relative humidity, it appears that a press room condition of 50 to 55 percent relative humidity is the desired design point, rather than either 42 percent or 60 to 65 percent. Obviously, if paper can be purchased with a moisture content of 5.5 percent there will be no need of hanging the paper. Then the film of moisture picked up by the paper from the offset blanket will have the least dimensional effect. Also, a 50 to 55 percent relative humidity condition will be far more comfortable than 60 to 65 percent during hot summer weather, assuming that the system has been designed primarily for humidity control (as it should be).

From the foregoing data, it would appear that the lithographer without some measure of humidity control is at a serious disadvantage, inasmuch as operating costs, service, and quality are again becoming important competitive factors.

In addition to the real value of humidity control in the press room, it has also been found that light sensitive lithographic press plates when dried in equilibrium with 50 to 55 percent relative humidity require about one-third less exposure time in the photo composing machine than do plates in equilibrium with 42 percent relative humidity. In the photo department, humidity control is also desirable especially on large film sizes. Since film is derived from cellulose (excluding vinyl) it is obvious that a 50 percent relative humidity at the time of exposure and at the time of use in either a vacuum frame or photo composing machine will keep variations in size down to a minimum, assuming that equilibrium conditions are attained in both instances.

Such other questions as the number of air changes and condensation control are subjects every qualified engineer can work out for any given location.

In conclusion, it appears that "D.S." has in effect indicated quite clearly that the lithographic industry is not of one mind as to what are the best process conditions, and perhaps the speaker he refers to was merely quoting others in the industry who believe that 42 percent relative humidity is the desired design point. As previously indicated, our experience with this problem points to a 50 to 55 percent relative humidity as the correct range.

We sincerely hope that you will hear from other readers who will give us the benefit of their views on this subject.—H.A.

COST OF A GOOD SYSTEM REASONABLE

THE LETTER from "D.S." in the January issue on air conditioning for printing plants poses a question and challenge to the air conditioning industry. It is my opinion that he is not only entitled to an answer from, and by, the industry but also that *his* industry is entitled to a reasonable economic response from ours.

Not to tear "D.S." down but, for a moment, let's look and see what is said. The end of his first paragraph states that "they have it, but we can't find it." But, later, he says in effect, "There aren't any intermediate air conditioning systems—only the finest." Hence, from these comments, I interpret "D.S.'s" troubles to have their origin in the economy (cost) of air conditioning rather than in the availability of equipment.

"D.S." says that there are only 10 to 25 days a year when an air conditioning system is really needed. Consequently, "D.S." is thinking in terms of a relatively short yearly usage and a comparatively heavy investment for the time he can use it. However, we all know, in the air conditioning industry, that once the customer has air conditioning on his premises, *he will use it almost every day*. And, he'll have a better product.

What does "D.S." say that he wants? Here are his words: "We believe that a humidity and temperature control with a reduction of 10 to 20 deg is sufficient." That is neither difficult nor expensive.

The simplest, cheapest, and most satisfactory control in the long run would be to have sufficiently deep coils supplied with chilled water passing through a mixing valve which is controlled by a humidistat, and with the temperature controlled by reheat.

The area under consideration, 70,000 sq ft, indicates a centrifugal installation of approximately 300 to 350 tons as an estimate. Using strategically located conditioners will substantially reduce the installation costs and improve operation.

It is my opinion that "D.S." could get a perfect installation which should produce satisfactory results in all respects for around $3.00 per sq ft of conditioned floor area which, when amortized over 10 years, becomes $0.30 per sq ft per yr. This does not seem excessive.

I have two suggestions to make. One is that "D.S." does not try to compromise with quality by being sold a cheap installation—it is more satisfactory in the long run to spend 10 or 15 percent more to get a good installation. Second, an installation of this scope and nature justifies

QUESTION OF THE MONTH

ASKS FOR INFORMATION ABOUT SERVICING AIR CONDITIONING

You are invited to answer the following question from "J.C." Suitable comment on this matter will be published, and paid for at regular rates (with additional payment for sketches). Address The Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2.

"I would appreciate having any information you can send me—or any comments that other readers might be willing to make—on the management and operation of service departments in the air conditioning field. I am taking over this type of work for a contractor, and would like advice on methods, personnel, etc."—J.C.

QUESTION OF THE MONTH

the retaining of a consulting engineer (which I am not) whose knowledge and skill will insure the desired results without unnecessary expense.

One last point to be offered: Lithographic prints can be made, as "D.S." points out, at 60 to 65 percent relative humidity. Granted. My point is that to maintain the register, the relative humidity must be maintained *evenly, as well as the temperature.*

"D.S." question was, Can I get a satisfactory installation for my lithographic plant at a reasonable cost? The answer to "D.S." is *yes,* particularly if every effort is made to remove all sources of outside heat gain through insulation, roof sprinkling, awnings, etc., and asphaltic-aluminum paint on the roof.

Not to "plug" my own line, but if we, as the air conditioning industry, don't satisfy the "D.S.'s", we'll lose a big segment of business.—S.M.

OUTDOOR AIR CAN OFTEN BE USED

REFERRING TO "D.S.'s" question in January on air conditioning his printing plant, the most important consideration is the locality in which he is operating.

In places like California, a dry bulb temperature of 76 F and a relative humidity of less than 60 percent can be maintained by using mostly outside air and removing only sensible heat from the air.

Sensible heat removal is relatively cheap compared with latent heat removal.

If latent heat removal is necessary, a direct expansion coil with the proper apparatus dew point will maintain the proper relative humidity within the limits prescribed by "D.S." with a simple temperature control. Under these conditions, a minimum amount of outside air will be most economical. Usually 15 cfm per person is more than adequate for ventilation.

The relative humidity in the winter season can be largely controlled by the proportion of outside air used—except in colder climates, where it is advisable to use a humidifier.

I submit the accompanying sketch as a sample installation employing a direct expansion coil and steam coil for all year air conditioning.—K.S.

VENTILATING A CLUB KITCHEN

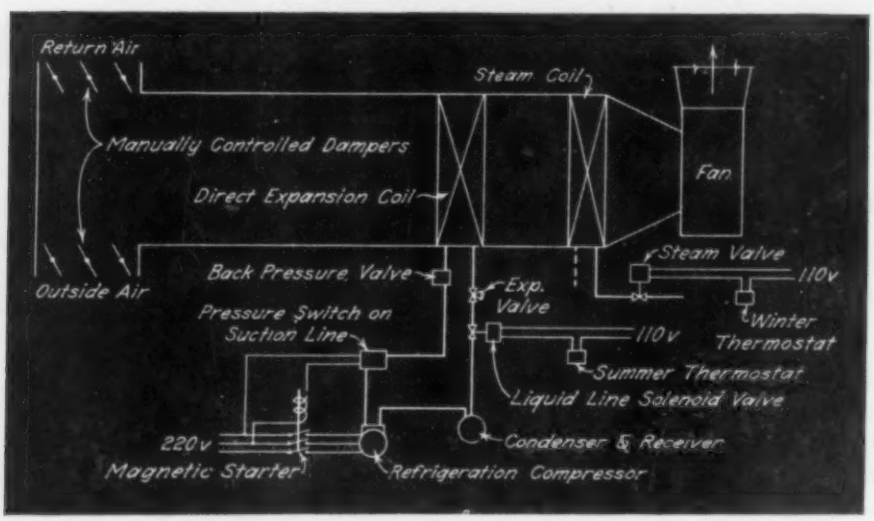
I WAS unable to comment on the Question of the Month regarding ventilating a club kitchen, as presented by "W.O.", when it was published in the November issue. However, since reading the comments by "R.H.R." and "K.R." in the January HPAC, I should like to make a few remarks.

Both "R.H.R." and "K.R." have very adequately discussed methods of removing air from the kitchen. However, I am not quite satisfied with the methods of inducing outside air into the kitchen. If the building is very tight and is used during weather when doors and windows are closed, the chef may experience some difficulty with his gas ranges and from fumes from the hot water boiler. To be maintained for proper operation of the ranges and boiler, the air to the kitchen must flow in at very nearly zero static pressure. "R.H.R." attempts to overcome this with an intake duct and filter. I question, though, the intake duct and filter resistance as compared to that of the flue.

I believe that this job would be more satisfactory with a positive method of air induction as well as exhaust. If this job is located in a section of the country where air washers operate satisfactorily, I should certainly consider this method of air induction. With a small amount of duct work, the air from the washer (evaporative cooler) can be discharged through jets or slots down over the main working area, thereby providing spot cooling. Filters could be installed either in the air washer or discharge duct from the air washer. An installation of this nature has been operating in Kansas City for a number of years and has been most satisfactory.

Some day, commercial kitchen help is going to discover that their jobs need not call for extreme discomfort.—R.W.U.

"K.S." submits this layout



OIL FUEL "RE-ENTERS COMPETITION"

The petroleum industry is re-entering a period of "typical American competition" with shortages overcome, production up, and rapidly expanding markets in view, R. M. Bartlett, vice president of Gulf Oil Corp., told oil men last month.

In an optimistic address before the semi-annual convention of the Connecticut Petroleum Industries Committee at Hartford, Mr. Bartlett reported that in the heavy fuel oil field, prices are down, stocks are up, and the industry is looking for new sales outlets. Mr. Bartlett commented:

"The industry's program of modernization and improvement has succeeded so well that we are again able to compete with coal . . . The battle of the two giant energy sources, oil and coal, is on again."

With reference to domestic heating oils, Mr. Bartlett said that 45 percent of American families express preference for central oil heat, but only 20 percent to date have oil fired equipment installed. "There are another 5,000,000 families who want oil heating," he said.

Heating, Piping & Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

Heating Panel Time Response Study

By A. B. Algren* and Ben Ciscel**, Minneapolis, Minn.

SUMMARY—In order to obtain basic information on automatic control of floor-panel heating systems, the authors made measurements of the transient conditions involved. A continuous record was made of water, ground, panel, air, and surface temperatures resulting from a sudden and large change in the heat input to the panel system.

THE problem of securing optimum comfort through automatic controls is often considered only after a house and its heating system are designed and built. Ordinarily such a heating system is designed or chosen on the basis of cost and its ability to supply the *steady-state* heat losses for the structure. The control problem is then a matter of selecting control equipment which is suitable to the rest of the installation, but which must also provide maximum comfort per dollar. The increasing number of homes being constructed with floor panel heating systems led to this study of the heat storage effects in the controlled system. In this study, heat input rates and losses were an important consideration.

*Professor of Heating, Ventilating, and Air Conditioning, University of Minnesota. Member of ASHVE.

**Control System Analyst, Minneapolis-Honeywell Research Division, Minneapolis-Honeywell Regulator Co.

Presented at the 55th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January 1949.

Experimental House

These tests were conducted in a small, well-insulated house (Fig. 1) without windows, and with only one door. The heating panel was a 4-in. concrete slab laid on 9 in. of crushed rock, with periphery insulation of rigid fiber board. The general arrangement and piping layout are shown in Fig. 2.

During the tests, no ventilation other than natural was used. The panel circulating water was heated by a heat exchanger connected to a steam system. The complete control system, as shown in Fig. 3, permitted considerable freedom in the choice of tests to be performed. The controller had two main sections: (1) for the hot water temperature, which depended upon the steam input to the heat exchanger; and (2) for determining the amount of hot water to be mixed with the recir-

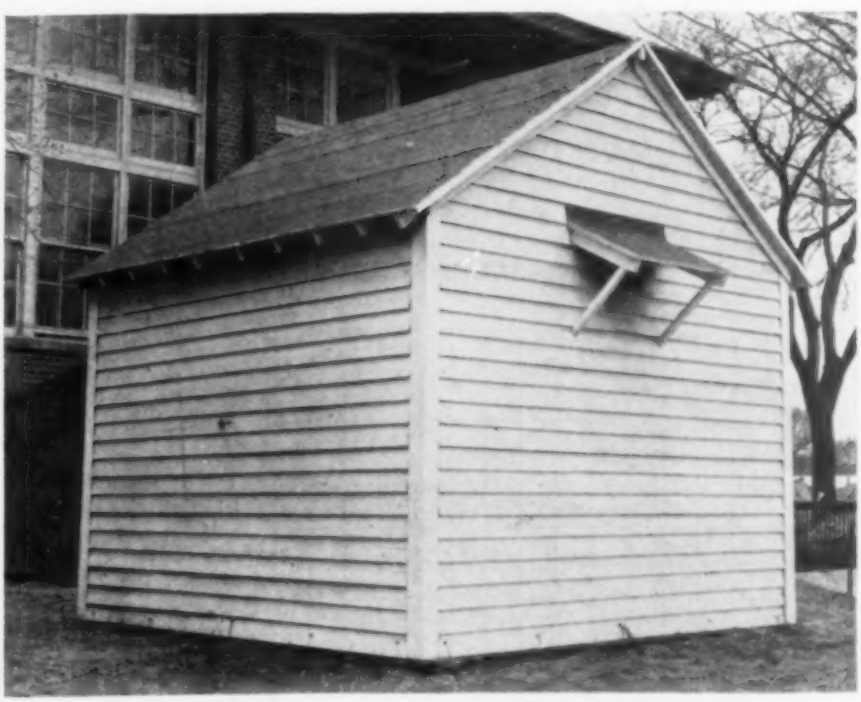


Fig. 1—Experimental house, University of Minnesota

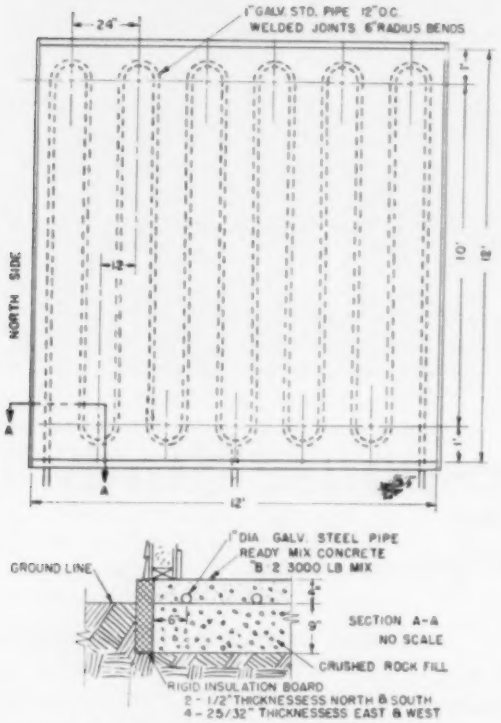


Fig. 2—Floor panel construction details

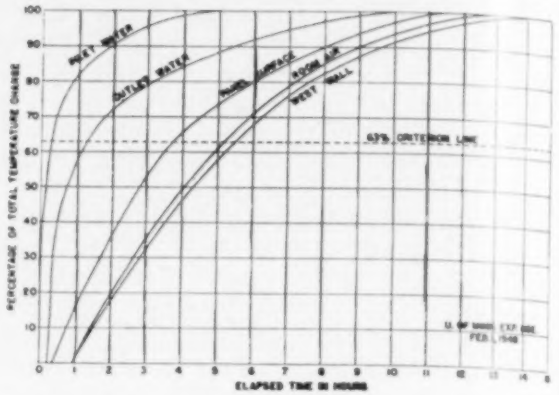


Fig. 5—Transient responses on a percentage basis

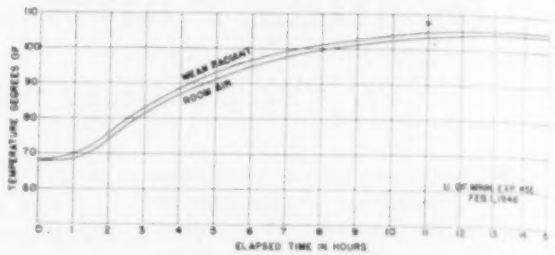


Fig. 6—Mean radiant and room air temperatures

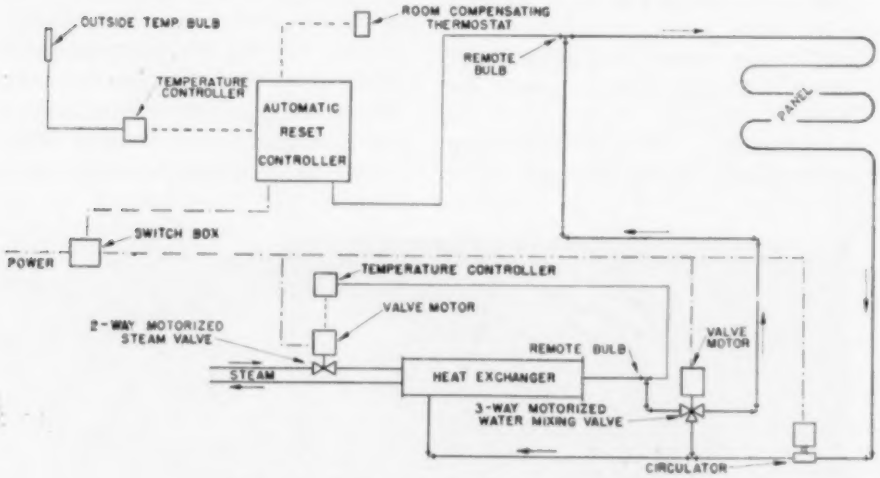


Fig. 3—Heating system with controls

culating water as a function of inside and outside temperatures.

More than a hundred points of temperature measurement, including the floor panel, were variously located inside the house. The room air temperature was measured by a system of wire thermocouples mounted about 4 in. from each wall and shielded from direct panel radiation. The average of these readings was taken as *room air temperature*. The *mean radiant temperature* was determined by networks of 12 thermocouples taped to each of the wall, ceiling and floor surfaces. The read-

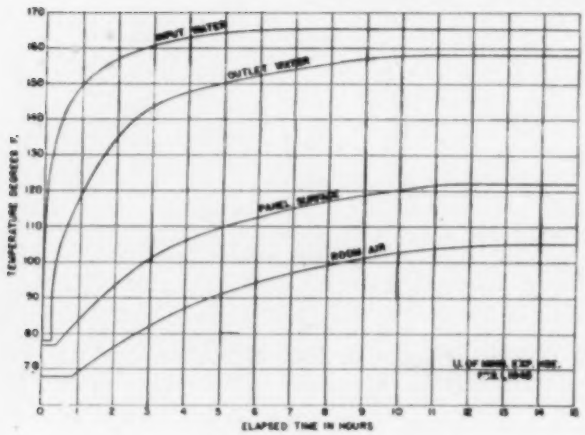


Fig. 4 *(Left)*—Transient responses within the house

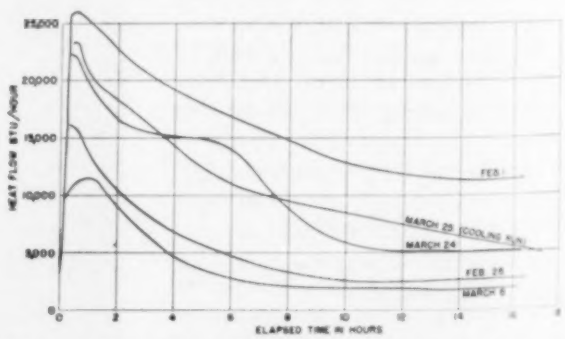


Fig. 7—Heat flow rates during transient conditions

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ing for each surface was weighted according to the area involved in calculating the mean radiant temperature.

Studies of random temperature variation in this and other houses led to the conclusion that ordinary control action produced temperature changes too small to be accurately traced in terms of time response. Hence, the investigators decided to produce large temperature changes artificially, in order that all the quantities involved would be measurable. Even though heat losses are not linear with mean temperature, it was considered better to have large readings with possible errors up to 5 percent rather than to have small readings difficult to interpret.

The transient conditions were initiated as follows. With the house under ordinary *steady-state* control, the inside controller was turned off and the recirculating valve was set for maximum hot water input to the panel. During the test, one or more recorders were used to record temperatures at a number of points of interest (up to 48 points, in some cases).

Transient Responses

The primary factor in house temperature control is the time lag of the heating panel. The authors were interested, however, in the transient conditions occurring in the *complete heating system*. Thus, the investigators began with the artificial demand for heat, and then measured water, panel, wall, and air temperatures. Some of these are shown in Fig. 4, with elapsed time as the independent variable.

In a study of the control problem, the actual values of temperature are not as important as the time required to reach the final (or some percentage of final) temperature. For that reason, the curves of Fig. 5 show the transient temperatures in terms of percentage change.

In the study of transient conditions, the times-of-response are often compared at the 63 percent mark; that is, where the temperature (or other variable) has gone through 63 percent of the difference between the initial and final values. Although these panel and house responses are more complicated than those produced by a single mathematical time constant, the responses can be compared at the 63 percent level. It will be noted in Fig. 5 that almost 4 hours were required by the panel to reach the 63 percent level.

The unique aspect of the data of Fig. 5 is not the panel lag, but in the hour and quarter lag of the room air behind the panel surface at the 63 percent criterion line. The west wall is included in Fig. 5 merely to indicate the trend of surface temperatures during the transient conditions studied. The surface temperatures of the other walls and of the ceiling were about the same as those for the west wall.

Temperature Measurements

The literature concerning comfort in terms of air and mean radiant temperature is extensive. For that reason, the experimental house was instrumented and constructed to facilitate measurements of surface temperatures and air temperatures.

The fact that the floor surface temperature was so far above room air temperature would lead most people to expect that the mean radiant temperature would be greater than the room air temperature in both time and magnitude. The curves of Fig. 6 show the actual temperatures for Feb. 1, 1948, in the well-insulated test house, where the mean radiant and room air temperatures differed by about one degree.

Tests on other days tended to verify the fact that mean radiant temperature (MRT) and air temperature T_a were generally within a degree of each other, even during transient conditions as sudden and as large as those obtained in these tests. This MRT did not take into account the temperature of doors and window areas. In a well insulated home having normal precentages of window and door area, however, the results would not be significantly affected by the temperature of these surfaces. Another point to be considered is the lack of ventilation, except that due to infiltration. This is being studied further for inclusion in a future report. In addition, this investigation is being expanded to include the effect of large window areas.

Heat Input Rates

In Fig. 5 the curves of inlet and outlet water temperatures vary widely from design values. Since water was constantly recirculated in the panel at a rate of 1.5 gallons per minute, the difference in temperature between inlet and outlet water was directly proportional to heat flow in Btuh.

Although the ordinary design calculations for a house of this type would indicate a design load of about 3000 to 5000 Btuh, the authors' transient heat inputs ran several times that amount. These flow rates are shown in Fig. 7, with the heat supplied to the panel appearing to be about six times that required to offset normal house losses. These large heat inputs could not be correlated directly with inlet water, outside air, or floor surface temperatures. However, the room air temperatures (see Fig. 8) rose in succeeding tests. This seems to indicate a change in the storage

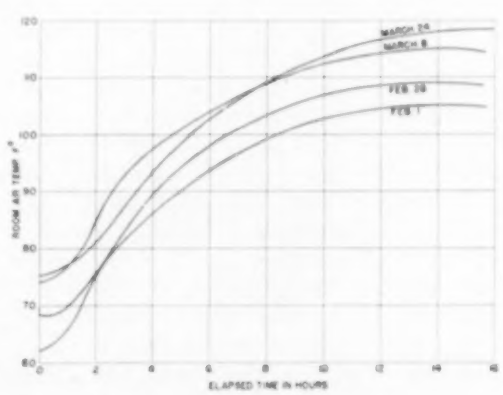

Fig. 8—Room air temperatures during transient conditions

JOURNAL SECTION

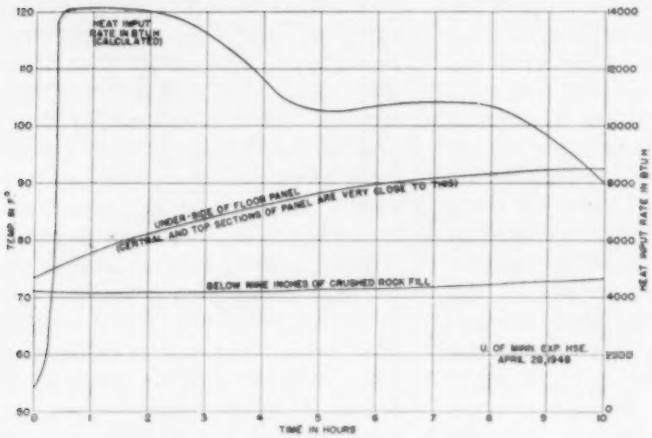
HEAT INPUT RATE IN BTUH (CALCULATED)
UNDER-SIDE OF FLOOR PANEL
(CENTRAL AND TOP SECTIONS OF PANEL ARE VERY CLOSE TO THIS)
BELOW NINE INCHES OF CRUSHED ROCK FILL
U. OF MINN. EXP. HSE.
APRIL 20, 1948
TEMP. IN °F
HEAT INPUT RATE IN BTUH
TIME IN HOURS

Fig. 9—Typical transient test conditions

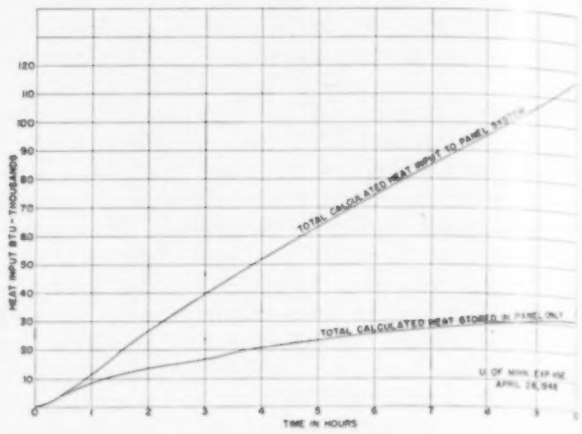
TOTAL CALCULATED HEAT INPUT TO PANEL SYSTEM
TOTAL CALCULATED HEAT STORED IN PANEL ONLY
HEAT INPUT BTU - THOUSANDS
TIME IN HOURS

Fig. 10—Time integral of heat input rates

or heat loss factor for the house after several heating cycles.

The trend of heat input rate (becoming progressively smaller) was reversed in the March 24 run. This reversal may have been due to a general thawing around the perimeter of the panel. (A study of temperatures on the preceding days showed a minimum temperature of –26 on March 11, although the average outside temperature for the week preceding the date of the test was +39 deg).

A cooling run is also shown (March 25, Fig. 7) giving an indication of heat flow back out of the panel after a heating run on the preceding day. This was accomplished by running cool water through the panels. The cooling run (Fig. 7) indicates that much of the heat input to the panel was stored below ground and was available for return depending upon the temperature level. However, much of this energy would probably have been lost in the earth if the cooling run had not been made.

The rate of heat input (shown in Fig. 7) brings up the question of total heat input to the panel system during a given period of time. For example, the data of Fig. 9 show a sudden heat input to the panel, with the calculated heat input rate reaching a maximum of about 14,000 Btuh. Also shown in Fig. 9 are the average temperatures at the underside of the floor panel and at the bottom of nine inches of crushed rock fill. The temperature at the bottom of the crushed rock fill (under the panel) showed little change even after 10 hours.

Attempting to account for these heat inputs, the authors calculated and plotted the storage in the panel (Fig. 10). The large difference between total heat stored in the panel and total heat supplied to the panel system indicated that panel storage accounted for only a part of the total heat input. The house thermal load accounted for about 2500 Btuh or a total of 25,000 Btu over the 10-hour period. About 50,000 Btu then remained to be accounted for by storage in the fill or structure, and by periphery losses. The need for insulation between the panel and the rock fill, as a means of improving the temperature response of the panel, is apparent.

At the end of 10 hours the total heat supplied to the panel system was about 116,000 Btu of which 25,000 had been transferred into the house. This indicates that if some system failure should cause a panel to be cooled, only a fraction of the heat supplied on resumption of heating would go into the house until the panel and fill came up to temperature.

Conclusions

1. The tests indicated that the mean radiant temperature was generally within a degree of the room air temperature.

2. The maximum air temperature rise rates were about 8 deg per hour and usually occurred about an hour after the heat input had been increased. Floor coverings or a smaller heat-input to the panel would further delay the rise of room air temperatures.

3. Heating plants for floor-panel heated houses should be capable of supplying heat well above the expected demand in order to take care of transient heat storage and losses.

4. Measurements taken before and after these periods of transient conditions show that *steady state* losses were within reasonable limits as contrasted with the excessive heat inputs required during heating up periods.

5. The air temperature was higher than the wall and ceiling temperatures. This indicated some heat transfer to these surfaces by convection and conduction as well as radiation.

6. The transient conditions in these tests were much larger than those which would be encountered in normal control operations, but the time lags involved should be independent of the temperature variation range.

Acknowledgments

The authors wish to extend their gratitude to the University of Minnesota and Prof. F. B. Rowley, director of the Engineering Experiment Station, University of Minnesota, for cooperation in making available the experimental house for these tests. Robert Atkinson and Don Vandergon worked as University Research Assistants on the project. Many engineers of the Minneapolis - Honeywell Regulator Co. contributed ideas which were helpful, but especial thanks are due to W. H. Kliever, director of research, and R. T. Squier, asst. director of research, for their guidance and assistance.

Psychrometric Factors in the Air Conditioning Estimate

By C. M. Ashley*, Syracuse, N. Y.

SUMMARY—This paper describes a method of estimating air conditioning load and selecting equipment. Elements considered in this method are: (1) *Sensible Heat Factor* of the room load; (2) Apparatus Dew Point; (3) *Bypass Factor*, a term for expressing coil performance in relation to the room air condition and the apparatus dew point; (4) Refrigerant side of the coil performance expressed in relation to the apparatus dew point; (5) Pre-selection of the coil depth and air velocity; (6) Consideration of *bypassed* air as part of room load.

The author compares his method with others and explains how he derived various factors such as the *Sensible Heat* and *Bypass Factors*.

CURRENT methods of estimating summer air conditioning load and selecting equipment have many elements in common, but differ in directness of approach and assurance of results. The present paper describes one method[1] used by the author for fifteen years, and compares it with other methods. It is hoped that such comparison will stimulate discussion which may establish a uniform method of estimating and of equipment selection.

A sample estimate form which presents an overall view of the method is included as Fig. 1.

Approach to the Estimate

The estimator of a heat load for summer air conditioning customarily starts with certain *design conditions*. These conditions are usually those for which system performance is to be guaranteed, and include:

a. Room air temperature and humidity.

b. Outside air temperature and humidity.

c. Number of people customarily in the space.

d. Internal sources of heat and moisture, such as lights, motors, cooking appliances, etc.

e. Ventilation standards to be maintained (usually in terms of cfm outside air per person).

Information must also be available as to the size of the room and the building construction. From the foregoing information the estimator determines (1) what heat and moisture load must be removed from the space to be conditioned; (2) what additional heat and moisture must be extracted from the ventilation air. Usually, the estimator also knows in advance the general type of air conditioning equipment to be used.

Sensible and Latent Heat

After determining the heat and moisture load of the room, the estimator next has to choose the temperature and humidity of the air to be supplied by the apparatus. He must also select the corresponding air quantity. The problem is illustrated graphically on the psychrometric chart (Fig. 2), where B represents the condition leaving the apparatus, and C the room condition. In order to reach room conditions, the air, as it passes from the apparatus to the room, must absorb the heat and moisture load liberated in the room. This is illustrated by line BC.

The first part of the heat gain of the air is associated with a change of temperature and is generally termed *sensible heat*. The sensible heat gain is a function of: (1) the temperature rise of the air moving from the apparatus to the room; (2) the rate of air flow; (3) the specific heat of the air (including its moisture).

The second part of the heat gain is associated with a change of moisture content of the air (humidity ratio) and is commonly known as *latent heat*. The latent heat gain is a function of: (1) the increase in moisture content of the air as it goes from apparatus to room conditions; (2) the rate of air flow; (3) the *latent heat* of the moisture per pound. The sum of the sensible heat and latent heat is commonly known as *total heat*. In Appendix I it is shown that for the range of conditions used in comfort air conditioning, it is possible to consider as constants both the *specific heat* of moist air and the *latent heat* per pound of dry air. The error in total heat should be under 1 percent. This is smaller than normally obtained by the use of uncorrected enthalpy at saturation.

Sensible Heat Factor

Since the same air is used for absorbing both moisture and *sensible* heat, the ratio of moisture to sensible heat gain per pound of dry air is the same as the ratio of the moisture load to sensible heat load to be absorbed in the room. The moisture content being the ordinate of Fig. 2 and the temperature the abscissa, their ratio is expressed by a constant slope, ABC. This ratio can be expressed either directly or in terms of related functions. A convenient related function is the ratio of sensible heat to total heat. This is called the Sensible Heat Factor.

$$(\text{SHF}) = \frac{Q_s}{Q_T} \quad \ldots\ldots (1)$$

where:

Q_s = Sensible heat load
Q_T = Total heat load

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* Chief Development Engineer, Carrier Corp. Member of ASHVE.

[1] Superscript numerals refer to References.

Presented at the 55th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January 1949.

On the psychrometric chart, the Sensible Heat Factor can be used to determine the slope of the line which joins the required state points of the supply air leaving the apparatus and that of the room air. (See Appendix I, Equation 12*a*). Since this line represents the locus of points satisfying the given ratio of moisture to sensible heat load, the condition of the air leaving the apparatus can be chosen anywhere along this line. Thus, a means is obtained for determining the supply air conditions required to give the *design* room conditions. Any leaving apparatus conditions not on this line cannot satisfy the design room conditions.

Apparatus Dew Point

In Fig. 2, the *apparatus dew point* is at point A, where the sloping Sensible Heat Factor line corresponding to Sensible Heat Factor joins the saturation line. It should be understood that the supply air condition need not be at A, but can be at any point (such as B) on the line AC.

Apparatus Dew Point Chart

Since the slope of a line on the psychrometric chart (Fig. 2) is difficult to determine accurately, an apparatus dew point chart of the alignment type has been devised (Fig. 3). This chart makes possible the reading of the *apparatus dew point* within ± 1 deg.

Coil Bypass Factor

In spray type dehumidifiers, the air leaving the dehumidifier is usually saturated and therefore is at the apparatus dew point. This simplifies the problem of choosing the correct apparatus dew point and air quantity.

When a coil is used as a dehumidifier, the air normally leaves in a condition less than saturated. This complicates matters, since the estimator must evaluate both the dry bulb temperature and the moisture content of the air leaving the coil. Moreover, these values are influeneed by the condition of the air entering the coil (which may not be at the room condition).

Referring to Fig. 2, if C represents the entering state point of the cooling coil and B the leaving state point, then the coil apparatus dew point is found by extending the straight line joining points C and B to the saturation line at A. It should be understood that this line is not necessarily a *condition line* for the air flow through the coil (Fig. 4). Rather, it represents the relationship between the entering and leaving terminal conditions. The slope of this line gives the Sensible Heat Factor for the coil.

For a very deep coil with entering and leaving state points on this line, the leaving condition would be practically at saturation. The *efficiency* of the coil could be considered as 100 percent, and the *inefficiency* zero percent. With a shallower coil, this no longer holds true, and the final state point will be at some point such as B of Fig. 2. In this case, it might be assumed that a portion of the air came in contact with the coil surface and was reduced to the condition of the coil apparatus dew point, while another portion passed through the coil without any change of condition[2]. The portion assumed to have passed unchanged through the coil is termed the *bypass* portion. It might be considered as the *inefficiency* of the coil. The remainder is termed the *dehumidified* portion.

The Bypass Factor (Appendix II, Equation 16) can be described as the ratio:

$$\frac{\text{leaving air temperature minus coil apparatus dew point}}{\text{entering air temperature minus coil apparatus dew point}}$$

Since the conditions of the air entering the coil, leaving the coil, and at the coil apparatus dew point, are on a straight line on the psychrometric chart, the Bypass Factor

Cooling Estimate

Fig. 1—Cooling Estimate Form

with respect to temperature and moisture will have the same value.

$$\text{Bypass Factor (BF)} = \frac{t_2 - t_s}{t_1 - t_s} = \frac{W_2 - W_s}{W_1 - W_s} \quad \ldots\ldots\ldots\ldots (2)$$

where: t_1, t_2 and t_s are the entering, leaving and apparatus dew point temperatures and W_1, W_2 and W_s are the corresponding humidity ratios.

Coil Performance, Air Side

The concepts of the coil Bypass Factor and of the coil *apparatus dew point* have been developed without any relation to the surface temperatures actually existing in the coil. Such a relationship is desirable in solving problems, and was implied in Carrier's 1911 statement of the laws of psychrometry[3]. Later, he[4] and others[5,6] developed the theory that in any process involving simultaneous transfer of heat and of moisture between air and a wet surface, the transfer of heat and of moisture between the surface and air is in proportion to the respective heads.

This relationship is presumably due to the diffusion of the water vapor molecules into the air at substantially the same rate as the transfer of heat (kinetic) energy between the molecules. (Actually, the water vapor diffusion is somewhat faster.) Since the rate of transfer of heat and moisture can be considered the same with respect to their heads, the condition of transfer is the same as if a portion of the air were brought in contact with the surface, and its temperature and moisture content were brought to the surface condition and then remixed with air passing unchanged through the coil.

But this is exactly the analogy used previously to illustrate the relationship between the entering and leaving coil conditions and the coil apparatus dew point. It can then be seen that the coil apparatus dew point has a physical as well as a mathematical significance. It is the uniform coil surface temperature necessary to produce the actual combination of entering and leaving conditions (as shown by line AC, Fig. 2).

In practice, the coil surface temperature is seldom uniform throughout, particularly where the coil has extended surface on the air side. A typical condition of the air as it passes through the coil is represented by the curved line CDB, Fig. 4, with initial surface temperature at E and the final surface temperature at F. However, the leaving air condition B is the same as though the coil surface temperature were constant at A, (See Appendix II).

It is thus possible to assign a single surface temperature as the composite of actual temperatures. Since this *effective* surface temperature is the lower terminus of the sloping line on which the entering and leaving coil conditions are located, it can be considered as the coil *apparatus dew point*.

Fig. 5 shows typical Bypass Factors for a line of finned coils. (The derivation of the coil Bypass Factor is shown in Appendix II). As will be seen, the Bypass Factor of a coil is a function primarily of the depth of the coil and, to a lesser extent, of the air velocity. It can be considered as a measure of the effective coil depth. The Bypass Factor thus indicates the air side performance of a chosen coil, and makes possible the calculation of the conditioned air quantity for a job for which the load and Sensible Heat Factor are known:

$$q = \frac{Q_s}{1.08(1\text{-BF})(t_r - t_s)} \quad \ldots\ldots\ldots\ldots (3)$$

where:

q = Flow of air, cfm

Q_s = Sensible heat load, Btu per hr

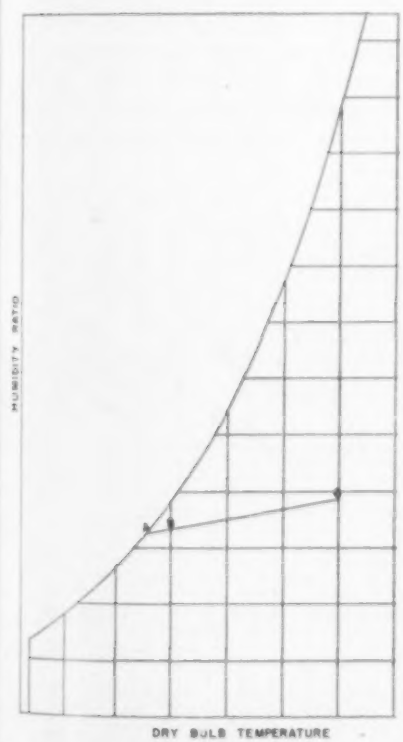


Fig. 2—Plot of Air Conditions on Psychrometric Chart

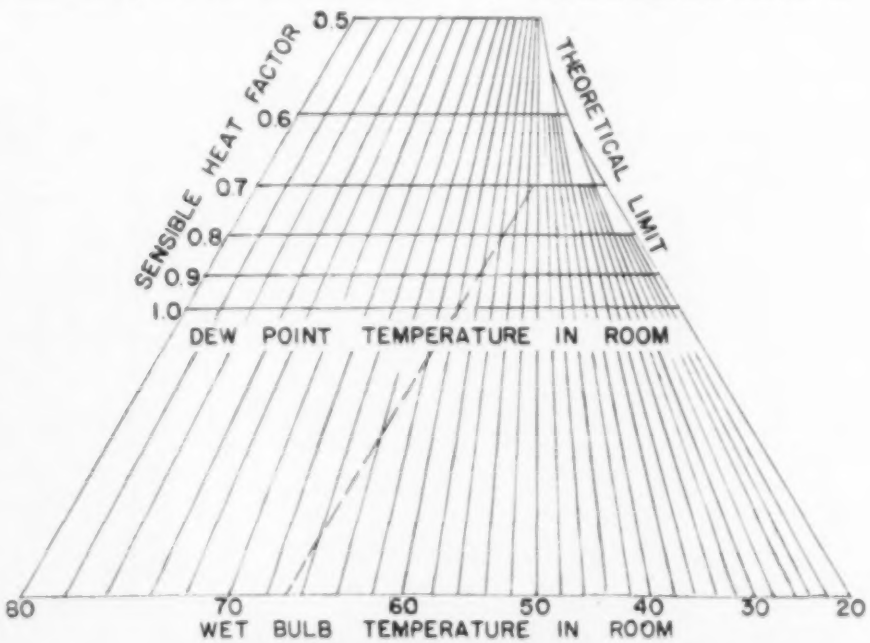


Fig. 3—Apparatus Dew Point Chart

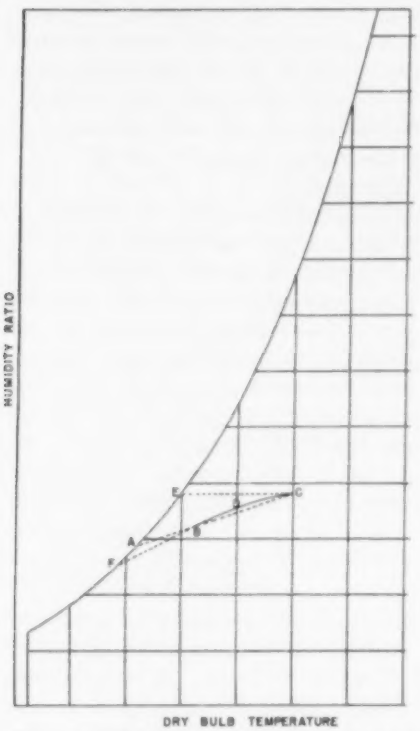


Fig. 4—Condition of Air Passing Through the Coil

BF = Bypass Factor
t_r = Room temperature
t_a = Apparatus dew point

Coil Performance, Overall

Since with all finned coil dehumidifiers there is a substantial difference between the coil *apparatus dew point* and the refrigerant temperature, it is essential to know the refrigerant side performance as well as the air side. Several methods of analyzing coil tests[11] and extending data to other coil conditions have been developed. The *total heat* or *three line* method developed by the author and by others[7] determines the terminal surface conditions for any chosen air and refrigerant conditions. This method (Fig. 6) is based upon determining the terminal surface conditions by making the ratio:

$$\frac{\text{total heat difference between air and surface}}{\text{temperature difference between surface and refrigerant}}$$

equal to

$$\frac{\text{heat transfer resistance of air film} \times \text{specific heat of moist air}}{\text{heat transfer resistance of refrigerant film and metal}}$$

From these data, the depth of the coil and hence the overall coil performance for the assigned conditions are determined. This basic method has been further refined[8].

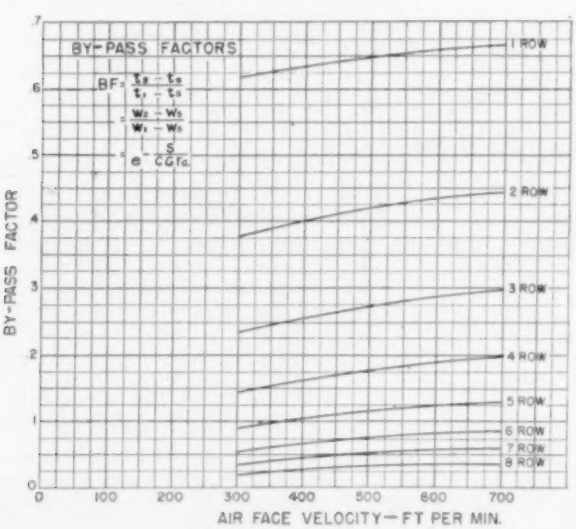
The refrigerant side heat transfer is expressed in terms of the temperature difference between the coil apparatus dew point calculated from the Bypass Factor (Equation 2) and the refrigerant temperature calculated as described previously. It has been found that this refrigerant side performance is relatively independent of the air side performance. For a given coil, using a non-volatile refrigerant, it may be expressed in terms of load and refrigerant flow[11]; for a volatile refrigerant, in terms of load[7]. (Illustrated in Table 1).

Problems Due to Outside Air

As far as possible, outside air for ventilation is brought through the apparatus, in order to reduce to a minimum the direct effect of outside air on the room conditions. This also increases the effectiveness of the air conditioning apparatus, because the entering air is increased in temperature and moisture content above the room conditions.

The ventilation requirements are determined by the sources of air contamination: people, food, etc. Thus, the outside ventilation air for a given job is chosen as a certain flow rate in cubic feet per minute. But the dehumidified air quantity is determined from the air conditions leaving the apparatus and depends upon the room load and the Sensible Heat Factor. Where a saturating type of dehumidifier is used, determining the condition of the air leaving the apparatus is simple, since the air is at saturation and is not affected by the condition of the air entering the coil.

On the other hand, where a coil dehumidifier is used which does not saturate the air, the air condition leaving the coil is a function not only of the effective coil surface temperature and of the coil Bypass Factor, but also of the entering air



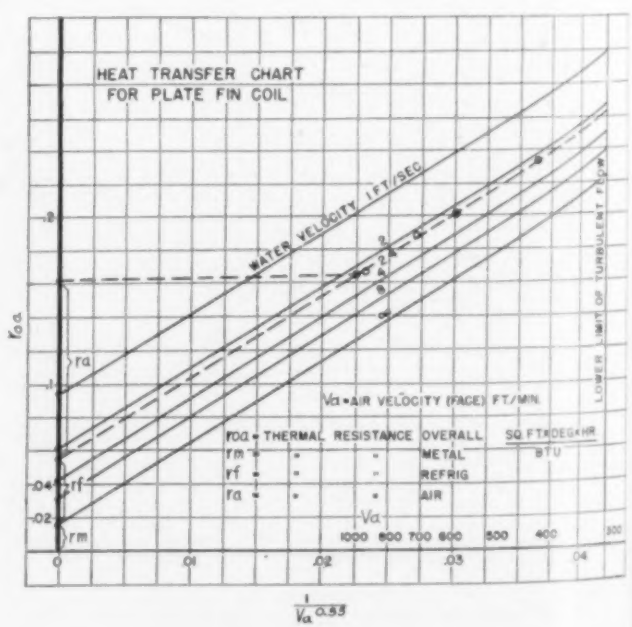

HEAT TRANSFER CHART
FOR PLATE FIN COIL
WATER VELOCITY LFT/SEC
Va = AIR VELOCITY (FACE) FT/MIN.
foa = THERMAL RESISTANCE OVERALL SQ FT DEG HR / BTU
rm = METAL
rf = REFRIG
ra = AIR
LOWER LIMIT OF TURBULENT FLOW

Fig. 5—Typical Bypass Factor for a Line of Finned Coils
Fig. 6 (Right)—Heat Transfer Chart for Plate Fin Coil

condition. But the entering air condition is unknown until the air quantity passing through the coil is chosen and the *percentage* of outside air can be determined.

From this dilemma there are three possible avenues of escape which do not involve cut-and-try or approximation: (1) choose the air velocity and depth, *i.e.* the *Bypass Factor*, of the dehumidifying coil; (2) choose the dehumidified air quantity; and (3) choose the leaving air conditions. The choice which is made is perhaps the most significant difference between estimating methods, as will be shown in a later section.

For the method here developed, the cooling coil is chosen as to air velocity and rows deep, as a part of the heat load calculation. The choice is simple and natural, because cooling coils are usually available in integral row increments of depth. This choice is based upon economic factors which are related to the type of job and type of load and which are readily learned by the estimator.

Since a portion of the air entering the coil is assumed to pass through without change of condition, this portion of the outside air can be considered as *bypassed* through the coil. The effect on the room load is the same as if this outside air entered the room independently of the apparatus, as infiltration air does. Thus, it is evident that the load due to the *bypassed* outside air can be treated in the same manner as infiltration load and therefore added to the room load.

The *dehumidified* portion of the outside air leaves at saturation, and its effect on the room is as though it were an equal amount of air from the room. Thus the new room sensible and latent heats (including the heat from the *bypassed* portion of the outside air) serve as the basis of a pseudo room Sensible Heat Factor line on which the coil apparatus dew point is located. The Bypass Factor, when applied to the room temperature minus the coil apparatus dew point, gives a pseudo air temperature leaving the apparatus. When used with the room sensible heat (including sensible heat from *bypassed* outside air) this pseudo air temperature permits the calculation of the dehumidified air quantity according to Equation 3.

The outside air load of the *dehumidified* or non-*bypassed* outside air must, of course, be added to the room total heat to give the grand

Table 1—Direct Expansion Cooling

GRAND TOTAL HEAT (Btu/hr in Thousands)						REFRIGERANT TEMPERATURES (F)													
COIL FACE AREA (SQ FT)						APPARATUS DEW POINT (F)													
2.25	4.5	6.75	9	13.5	18														
cubic feet per minute																			
1000	2000	3000	4000	6000	8000	38	40	42	44	46	48	50	52	54	56	58	60	62	64
4 ROWS BYPASS FACTOR 0.17																			
30.0	59.6	90.5	122.5	182.0	242.0	*27*	*29*	*31*	33	35	37	39	41	43	45	47	49	51	53
32.0	63.6	95.0	131.0	194.0	258.0	*26*	*28*	*30*	32	34	36	39	41	43	45	47	49	51	53
34.0	67.6	103.0	139.0	206.0	274.0	*26*	*28*	*30*	32	34	36	38	40	42	45	47	49	51	53
36.0	71.5	109.0	147.0	218.0	290.0	*26*	*28*	*30*	32	34	36	38	40	42	44	47	49	51	53
38.0	75.5	115.0	155.0	231.0	306.0	*26*	*28*	*30*	32	34	36	38	40	42	44	46	48	51	53
40.0	79.5	121.0	163.0	243.0	322.0	*25*	*27*	*29*	*31*	33	36	38	40	42	44	46	48	50	53
42.0	83.5	127.0	171.5	255.0	338.0		*27*	*29*	*31*	33	35	38	40	42	44	46	48	50	52
44.0	87.4	133.0	179.5	267.0	355.0		*27*	*29*	*31*	33	35	37	49	42	44	46	48	50	52
46.0	91.4	139.0	188.0	279.0	371.0		*26*	*28*	*30*	33	35	37	39	41	44	46	48	50	52
48.0	95.5	145.0	196.0	291.0	387.0		*26*	*28*	*30*	32	34	37	39	41	43	45	48	50	52
50.0	99.4	151.0	204.0	303.0	404.0		*25*	*27*	*30*	32	34	36	38	41	43	45	47	49	52
52.0	103.2	157.0	212.0	316.0	419.0			*27*	*29*	*31*	34	36	38	40	43	45	47	49	51
54.0	107.2	163.0	220.0	328.0	436.0			*26*	*28*	*31*	33	35	38	40	42	44	47	49	51
56.0	111.0	169.0	228.0	340.0	452.0			*26*	*28*	*30*	33	35	37	39	42	44	46	48	51
58.0	115.5	176.0	237.0	352.0	468.0			*25*	*27*	*30*	32	34	37	39	42	44	46	48	50
60.0	119.0	181.0	245.0	364.0	484.0				*26*	*29*	*31*	34	36	38	41	43	45	48	50
62.0	123.2	187.0	253.0	376.0	500.0				*26*	*28*	*31*	33	36	38	40	43	45	47	49
64.0	127.0	193.0	260.0	388.0	516.0				*25*	*28*	*30*	33	35	38	40	42	45	47	49
66.0	131.0	199.0	268.0	400.0	532.0					*27*	*29*	32	35	38	40	42	45	47	49
68.0	135.2	205.0	277.0	413.0	548.0					*26*	*28*	*31*	34	36	39	41	43	46	49
70.0	139.2	212.0	285.0	425.0	565.0					*25*	*27*	*30*	33	35	38	40	43	45	49
6 ROWS BYPASS FACTOR 0.07																			
36.0	71.5	109.0	147.0	218.0	290.0	*30*	32	34	36	38	40	42	44	46	48	50	52	54	56
38.0	75.5	115.0	155.0	231.0	306.0	*29*	*31*	33	36	38	40	42	44	46	48	50	52	54	56
40.0	79.5	121.0	163.0	243.0	322.0	*29*	*31*	33	35	38	40	42	44	46	48	50	52	54	56
42.0	83.5	127.0	171.5	255.0	338.0	*29*	*31*	33	35	37	39	41	44	46	48	50	52	54	56
44.0	87.4	133.0	179.5	267.0	355.0	*28*	*31*	33	35	37	39	41	43	45	48	50	52	54	56
46.0	91.4	139.0	188.0	279.0	371.0	*28*	*30*	32	35	37	39	41	43	45	47	49	52	54	56
48.0	95.5	145.0	196.0	291.0	387.0	*28*	*30*	32	34	36	39	41	43	45	47	49	51	53	56
50.0	99.4	151.0	204.0	303.0	404.0	*27*	*29*	32	34	36	38	40	43	45	47	49	51	53	55
52.0	103.2	157.0	212.0	316.0	419.0	*27*	*29*	*31*	34	36	38	40	42	44	47	49	51	53	55
54.0	107.2	163.0	220.0	328.0	436.0	*26*	*29*	*31*	33	35	38	40	42	44	46	49	51	53	55
56.0	111.0	169.0	228.0	340.0	452.0	*25*	*28*	*30*	33	35	37	39	42	44	46	48	50	53	55
58.0	115.5	176.0	237.0	352.0	468.0		*27*	*30*	32	35	37	39	41	43	46	48	50	52	55
60.0	119.0	181.0	245.0	364.0	484.0		*27*	*29*	32	34	36	39	41	43	45	48	50	52	54
62.0	123.2	187.0	253.0	376.0	500.0		*26*	*29*	*31*	33	36	38	40	43	45	47	50	52	54
64.0	127.0	193.0	260.0	388.0	516.0		*25*	*28*	*30*	33	35	38	40	42	45	47	49	52	54
66.0	131.0	199.0	268.0	400.0	532.0			*25*	*27*	*30*	32	35	37	39	42	44	47	49	51	54
68.0	135.2	205.0	277.0	413.0	548.0				*26*	*29*	*31*	34	36	39	41	44	46	49	51	53
70.0	139.2	212.0	285.0	425.0	565.0				*25*	*28*	*30*	33	36	38	41	43	46	48	51	53

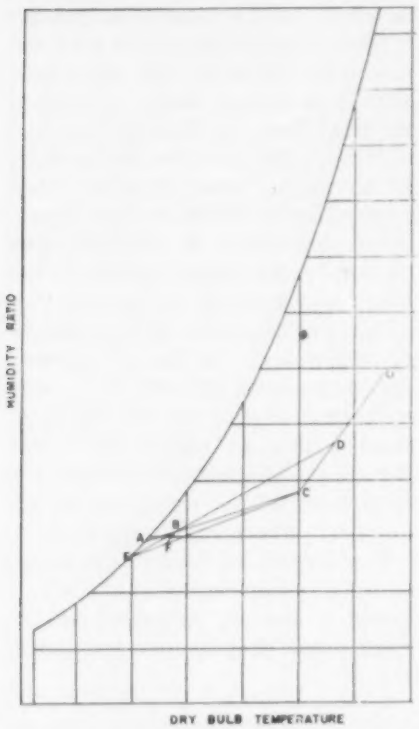


Fig. 7—Relation of Outside Air Load and Bypassed Air Load on Psychrometric Chart

total heat. It should be noted that when this method is used it is not necessary to determine the actual air conditions entering and leaving the coil.

The relationships described can be seen graphically on the psychrometric chart, Fig. 7. The slope of the line AC represents the Sensible Heat Factor for the room condition at C and A the apparatus dew point for the room. Let O be the outside air condition and D on line CO the condition of the mixed air entering the coil. Assume the condition of the air leaving the coil as B on line AC. Then the coil apparatus dew point must be at E where the extension of the line DB intersects the saturation line and this must be the *surface temperature* of the coil. This corresponds to a room Sensible Heat Factor represented by the line EC.

If now a line is drawn from the leaving coil condition B parallel to CO meeting line EC at F, two similar triangles EBF and EDC are formed. Therefore, EB/ED = BF/DC = EF/EC = coil Bypass Factor (equivalent sides of similar triangles). But since DC represents the outside air load, BF must represent the *bypassed* portion of the outside air load. This can be attached to BC representing the room load to give a new equivalent or pseudo room load represented by FC. By means of the transfer to the room load of the *bypassed* portion of the outside air load (represented by BF) a new value can be determined for room Sensible Heat Factor (represented by line FC). Thus the coil apparatus dew point E becomes the pseudo room apparatus dew point. The air quantity passing through the coil may thus be figured from the coil apparatus dew point E, the room condition C, and the coil Bypass Factor EF/EC.

Calculating Air Conditioning Estimate

A typical cooling load estimate made in accordance with the principles outlined in this paper is shown on the cooling estimate form (Fig. 1). The highlights of the estimate are: (1) pre-selection of the outside and room conditions; (2) calculation of the ventilation requirements in terms of air quantity; (3) determination of the *bypass* outside air load using the coil *Bypass Factor* determined from the preselected coil depth and face velocity; (4) determination of the Sensible Heat Factor and from this the coil apparatus dew point; (5) calculation of the dehumidified air quantity based upon the room conditions, the coil apparatus dew point and the coil *Bypass Factor*; and (6) selection of the equipment as determined by the grand total heat and the refrigerant temperature.

Comparison of Methods

Most load estimating and equipment selection methods which are designed to produce a guaranteed result recognize the need for breaking down the load into its sensible and latent components. These methods use some equivalent of the Sensible Heat Factor for determining the state point of the air leaving the apparatus. Some of these systems also recognize the principle of the coil apparatus dew point and of the Bypass Factor.

The principal differences occur in the determination of the dehumidified air quantity and the selection of the cooling coil. Table 2 illustrates in chart form the salient differences between the method presented in this paper and other possible means of treating the problem.

Table 2—Comparison of Different Systems of Load Estimating and Cooling Coil Selection

Item	SYSTEMS				
	1[c]	2	3	4	5
Outside Air, cfm	Given	Given	Given	Given	Given
Outside Air — D.B.	Given	Given	Given	Given	Given
Outside Air — Moist. Cont.	Given	Given	Given	Given	Given
Room D.B.	Given	Given	Given	Given	Given[a]
Room Moist. Cont.	Given	Given	Given	Given[a]	Given[a]
Room — Sensible Heat	Calculate[b]	Calculate	Calculate	Calculate	Calculate[a]
Room — Latent Heat	Calculate[b]	Calculate	Calculate	Calculate[a]	Calculate[a]
Outside Air — Sens. Heat.	Calculate[b]	Calculate	Calculate	Calculate	Calculate[a]
Outside Air — Latent Heat.	Calculate[b]	Calculate	Calculate	Calculate[a]	Calculate[a]
Sensible Heat Factor	Calculate	Calculate	Calculate	Calculate[a]	Calculate[a]
Apparatus Dew Point	Result 2	Result 5	Result 5	Assume[a]	Assume[a]
Apparatus cfm	Result 3	Assume	Result 1	Result 1	Result 1
Apparatus — D.B.	—	Result 1	Assume	Assume	Assume
Apparatus — Moist. Cont.	—	Result 2	Result 2	Assume[a]	Assume
Outside Air & Room Mixture — D.B.	—	Result 3	Result 3	Result 2	Result 2
Outside Air & Room Mixture — Moist. Cont.	—	Result 4	Result 4	Result 3	Result 3
Coil — Velocity	Assume	Assume	Assume	Assume	Assume
Coil — Depth	Assume	Result 7	Result 7	Assume	Result 4
Coil — Bypass Factor	Result 1	Result 6	Result 6	Assume[a]	Result 5
Refrigeration Temp.	Result 4	Result 8	Result 8	Result 4	Assume

Notes: Numbers indicate normal sequence of resultant steps.
[a] Approximate only.
[b] *Bypassed* outside air heat included in room heat.
[c] No. 1 system is that described in this paper.

Systems 1, 2 and 3 represent the three solutions listed under *Problems Due to the Outside Air*. In System 4, the coil depth and the coil leaving air conditions are assumed, whereas in System 5 the refrigerant temperature and the coil leaving air conditions are assumed. Both systems may result in substantial deviations, from design, of the room humidity and of the latent heat load. The numbers following the results indicate the probable order of the steps and provide some index to the calculation.

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Symbols

A = Face area of the coil, square feet.
(BF) = Bypass Factor.
c = Specific heat of moist air, Btu per pound. (dry air)
G = Weight flow of air, pound per hour. (dry air)
h = Enthalpy of moist air, Btu per pound. (dry air)
K = Mass transfer coefficient of moisture through air film, pounds per square foot × hours × pounds per pound.
L = Coil characteristic ratio. (See Equation 30)
M = Moisture transfer, pounds per hour.
q = Volume flow of air, cubic feet per minute. (standard conditions)
Q_L = Latent heat, Btu per hour.
Q_s = Sensible heat, Btu per hour.
Q_T = Total heat, Btu per hour.
r_a = Heat flow resistance of air film, hours × square feet × degree per Btu.
r_m = Heat flow resistance of metal, hours × square feet × degree per Btu.
r_r = Heat flow resistance of refrigerant film, hours × square feet × degree per Btu.
S = Surface area of the coil (air side), square feet.
(SHF) = Sensible Heat Factor.
t = Temperature, Fahrenheit degree.
t_s = Coil apparatus dew point.
V_s = Velocity of the air over the coil face, feet per minute.
W = Humidity ratio, pounds of moisture per pound of dry air.
ρ = Density, pounds per cubic foot.

Subscripts

$_1$ = Entering condition
$_2$ = Leaving condition
$_r$ = Room condition
$_s$ = Surface condition
$_w$ = Moisture condition
$_r$ = Refrigerant condition

APPENDIX I

Derivation of Sensible Heat, Latent Heat, Total Heat and Sensible Heat Factor

The enthalpy of moist air, h, may be expressed within the limits of 32 to 100 F with an accuracy of 0.10 percent by the equation:

$$h = (0.2402 + 0.44W)t + 1061W \quad \ldots\ldots\ldots\ldots (4)$$

In a process involving the removal of heat and of moisture from air, the heat removed per unit weight of dry air is equal to the difference of the enthalpies corresponding to the entering and leaving conditions of the air minus the enthalpy of liquid water condensed from a unit weight of dry air. For processes above 32 F, this heat difference may be expressed as:

$$\frac{Q_T}{G} = h_1 - h_2 - (t_{w2} - 32)(W_1 - W_2) \quad \ldots\ldots\ldots\ldots (5)$$

Substituting Equation 4 in Equation 5 and re-arranging:

$$\frac{Q_T}{G} = (0.2402 + 0.44W_2)(t_1 - t_2) + (1093 - t_{w2} + 0.44t_1)(W_1 - W_2) \quad \ldots\ldots\ldots\ldots (6)$$

It is possible to simplify Equation 6 by assuming certain specific values for some of the terms of second order value. Substitute representative values for these second order terms as follows:

W_2 = 0.0087 lb moisture per pound of dry air (53½ F dew point)
t_1 = 80 Fahrenheit
t_{w2} = 52 Fahrenheit

Then:

$$\frac{Q_T}{G} = 0.244\ (t_1 - t_2) + 1076\ (W_1 - W_2) \quad \ldots\ldots (7)$$

If other values of W_2, t_1 and t_{w2} are substituted in Equation 6 and compared with Equation 7, then Equation 7 can be used in solving the normal problem of cooling and removing moisture from room air for comfort conditioning purposes, with an error varying from 0 to ½ percent; whereas the same equation can be used for determining the removal of heat and moisture from outside air with an error of ½ to 1 percent.

The first term on the right side of Equation 7 may be described as the *sensible heat*, either based on a unit weight or on the pounds per hour of air used in the process, as in Equation 8. Or it may be expressed in terms of the volume of standard air in cubic feet per minute as in Equation 8*a*:

$$Q_s = 0.244G\ (t_1 - t_2) \quad \ldots\ldots\ldots (8)$$

$$Q_s = 1.08q\ (t_1 - t_2) \quad \ldots\ldots\ldots (8a)$$

Note that the specific volume of standard air (0.075 pounds per cubic foot) must be multiplied by the volume ratio of dry to moist air at the assumed condition $\left(1 - \frac{0.0087}{0.62}\right) = 0.986$ to get the specific volume of dry air at standard conditions. Similarly, the second term of Equation 7 may be described as *latent heat* and may be expressed either in terms of unit-weight or of pounds of air per hour as in Equation 9, and may also be expressed in terms of standard air (cfm) as in Equation 9*a*:

$$Q_L = 1076G\ (W_1 - W_2) \quad \ldots\ldots (9)$$

$$Q_L = [0.68q]\ [7000\ (W_1 - W_2)] \quad \ldots\ldots\ldots\ldots (9a)$$

Note that the multiplying factors in Equations 8*a* and 9*a* are those used in

connection with the air conditioning estimate for sensible heat and latent heat respectively; also, that the second bracketed term of Equation 9a includes the conversion factor from pounds to grains. If W were in grains per pound, the 7000 would be cancelled out. Q_T, the total heat, may then be expressed as the sum of the sensible and latent heat:

$$Q_T = Q_s + Q_L \quad \ldots\ldots\ldots\ldots (10)$$

While the terms *sensible*, *latent*, and *total* heat do not have an absolute physical significance, they are, in practice, very significant to the air conditioning engineer, and their use greatly simplifies the mechanics of load computation.

Sensible Heat Factor is defined as the ratio of the sensible to the total heat:

$$(\text{SHF}) = \frac{Q_s}{Q_T} \quad \ldots\ldots\ldots\ldots (11)$$

By substituting Equations 8 and 9 in Equation 11, Sensible Heat Factor may be expressed in terms of temperature difference and humidity ratio difference:

$$(\text{SHF}) = \frac{0.244\ (t_1 - t_2)}{0.244\ (t_1 - t_2) + 1076\ (W_1 - W_2)} \quad \ldots\ldots (12)$$

This expression can be simplified further to show the humidity ratio difference to the temperature difference as a simple ratio:

$$(\text{SHF}) = \frac{1}{1 + 4400\ \dfrac{(W_1 - W_2)}{(t_1 - t_2)}} \quad \ldots\ldots\ldots\ldots (12a)$$

Equation (12a) provides the information necessary to plot the slope of the condition line corresponding to any sensible heat factor on the psychrometric chart. Note that W in Equation 12a is in pounds per pound.

John R. Allen Memorial Library

The ASHVE Committee on Research gratefully acknowledges the following gifts to the John R. Allen Memorial Library, which is maintained at the Society's Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio.

Industrial Hygiene and Toxicology, Vol. I, edited by Frank A. Patty. Gift of W. N. Witheridge.

Refrigeration and Air Conditioning, by Richard C. Jordan and Gayle B. Priester. Gift of the authors.

The presentation of textbooks and pamphlets in the field of heating, ventilating, and air conditioning and allied fields, by authors and others, will be welcomed by this library. Suitable book plates carry the name of the donor and date of the gift.

APPENDIX II

Derivation of Bypass Factor

The sensible cooling capacity Q_s of a coil may be stated in terms of its surface, air film resistance, and log mean temperature difference as:

$$Q_s = \frac{S}{r_s} \times \frac{(t_1 - t_s) - (t_2 - t_s)}{\log_e \left(\dfrac{t_1 - t_s}{t_2 - t_s} \right)} \quad \ldots\ldots\ldots\ldots (13)$$

where t_s is assumed to be constant. But Q_s can also be expressed as:

$$Q_s = Gc\,(t_1 - t_2) = 60 V_s A \rho c\,(t_1 - t_2) \quad \ldots\ldots (14)$$

Combining Equations 13 and 14 and simplifying:

$$Gc = \frac{S}{r_s} \times \frac{1}{\log_e \left(\dfrac{t_1 - t_s}{t_2 - t_s} \right)} \quad \ldots\ldots\ldots\ldots (15)$$

This may be converted from the log form to a power of e in terms of the Bypass Factor (BF):

$$\text{BF} = \frac{t_2 - t_s}{t_1 - t_s} = e^{-\frac{S}{Gcr_s}} = e^{-\frac{S}{60 V_s \rho c r_s}} \quad \ldots\ldots\ldots\ldots (16)$$

Lewis[5] has developed in approximate form the relationship of the heat transfer from air to surface and the corresponding mass transfer of moisture as:

$$cr_s K = 1 \quad \ldots\ldots\ldots\ldots (17)$$

In a manner similar to Equation 13 for heat transfer, the mass transfer of moisture to the coil can be written:

$$M = SK \times \frac{(W_1 - W_s) \times (W_2 - W_s)}{\log_e \dfrac{W_1 - W_s}{W_2 - W_s}} \quad \ldots\ldots\ldots\ldots (18)$$

where

W_s is assumed constant.

Likewise:

$$M = G(W_1 - W_2) = 60\ V_s A \rho (W_1 - W_2) \quad \ldots\ldots (19)$$

Combining Equations 17, 18 and 19 and recasting as in Equations 15 and 16:

$$\text{BF} = \frac{W_2 - W_s}{W_1 - W_s} = e^{-\frac{S}{Gcr_s}} = e^{-\frac{S}{60 V_s \rho c r_s}} \quad \ldots\ldots\ldots\ldots (20)$$

In the given equations t_s and W_s were assumed constant but Equations 13 and 14 can also be written:

$$\sum_1^2 \Delta Q_s = r_s \sum_1^2 (t - t_s)\ \Delta S = Gc\,(t_1 - t_2) \quad \ldots\ldots (21)$$

where values of $(t - t_s)$ are the actual values of air to surface temperature difference through the coil as shown in Fig. 5. From Equations 13 and 21 the mean temperature difference is:

$$\frac{(t_1 - t_s) - (t_2 - t_s)}{\log_e \left(\dfrac{t_1 - t_s}{t_2 - t_s} \right)} = \frac{1}{S} \sum_1^2 (t - t_s)\ \Delta S \quad \ldots\ldots (22)$$

It is apparent that the left-hand term of Equation 22 is a particular form of the right-hand term in which t_s is given a value to satisfy Equation 13. Thus, the significance of t_s or coil apparatus dew point is that it is the constant surface temperature necessary to produce the same results as were actually produced with the variable surface temperature. The same reasoning applies to the mass transfer of moisture, and the constant surface condition can be considered the same for both heat and moisture transfer.

1948 Annual Report of the Committee on Research

by L. N. Hunter

THERE were several noteworthy events in the Society's research activities for 1948. At the June meeting, the Society amended the regulations governing the Committee on Research so that research accounting and payment of bills for research expenditures could be done at the Laboratory instead of at the New York office. The accounting books for research were transferred to the Laboratory on July 1.

One of the important highlights was the trip to England by the Director of Research on behalf of the Society, to review the research that is being carried on there and to establish a channel for the exchange of research reports between the two countries. The Director's report of this trip contains considerable information of value to the Committee on Research and to the Technical Advisory Committees.

The Committee on Research wishes to acknowledge the splendid cooperation and sympathetic consideration given by Council to research matters. The Finance Committee of Council has been especially helpful on financial and budget problems. The Committee also wishes to express appreciation to the officers of the Society for valuable help and suggestions given on research matters throughout the year.

Committee Activities

The Committee on Research held four meetings during the year, three of them being full-day meetings. The Chairman wishes to thank the members of the Committee for their excellent cooperation and record of attendance at these meetings, and for the seriousness with which they have assumed their responsibilities.

Altogether there were 21 Technical Advisory Committee meetings held during the year. A number of these Committees met several times. Many of the meetings were held in Cleveland, at the Laboratory, which has excellent facilities for holding such meetings. Another advantage in holding the meetings at the Laboratory is that the members of the staff are given an opportunity to participate in the meetings and to have personal contact with the Committee members. The total committee attendance at these meetings was 166 members and 101 guests, including Laboratory staff members.

The Technical Advisory Committees play an important part in our research organization in making recommendations on our research program, and suggesting authors and titles for papers and rendering assistance in the preparation of material for THE GUIDE. Their support and cooperation are appreciated, especially that of the Committee Chairmen, on whom the success of the Committee's activities depend to a large extent. It is planned to rely to an even greater extent on the Technical Advisory Committees for help in the future.

The formation of two new Technical Advisory Committees was authorized by the Committee on Research, one on Odors and one on the Heat Pump.

Support by Industry

The Committee wishes to acknowledge the splendid financial support received from industry during this past year. The number of organizations throughout the industry which participated financially increased greatly and during 1948 numbered 242. A large share of the credit for stimulating industry's interest and desire to participate in our research work is due to the activities of C. A. McKeeman, assistant to the president, and to the work of the Committee on the Promotion of Research under the Chairmanship of J. E. Haines.

Research Policy

The Committee has discussed, on several occasions during this past year, the importance of determining the attitude and feelings of the Society membership towards our research program and its objectives. Consequently, the Committee requested that a letter be sent to the membership asking each member to express his views on the research activities of the Society. Replies to this letter are now being received. Those members who have not taken advantage of this opportunity for expressing their opinions are urged to do so immediately.

The Committee wishes to express appreciation for the continued interest of the membership in our research work. Many helpful suggestions have been received from members who are not on the Committee on Research or Technical Advisory Committees.

Council has urged the Committee on Research to place greater emphasis on physiological research and cooperative research, and to study the possibilities of closer cooperation with governmental agencies. The Committee has accepted the suggestion and has increased its expenditures for physiological research, planning further increases for 1949. During this past year a contract was negotiated with the Office of Naval Research involving the Laboratory's undertaking the design, development, and construction of a calorimeter for the rapid measurement and recording of heat exchange between the human body and its environment.

Universities, apparently, are mastering some of their post-war problems and now are paying more attention to fields of research, and many proposals involving cooperative research with the Society have been submitted during this past year. Some of these proposals have been accepted and the work is already under way. A number of others have been referred to the proper Technical Advisory Committees for their study and recommendations.

Publications and Papers

The publication policy of the Society as it affects research has been discussed by a special Committee on Publication Policy appointed by the President. According to the policy adopted, each

Presented at the 55th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January 1949.

L. N. Hunter

1948 Committee on Research

L. N. HUNTER, *Chairman*
R. D. MADISON, *Vice Chairman* A. C. FIELDNER, *Ex-Officio*
CYRIL TASKER, *Director of Research*

Executive Committee: L. N. Hunter, *Chairman;* R. D. Madison, *Vice Chairman;* L. G. Miller, L. P. Saunders, W. E. Zieber.

Three Years: A. B. ALGREN, M. C. GIANNINI, C. W. JOHNSON, E. R. QUEER, W. E. ZIEBER.

Two Years: R. C. CROSS, M. K. FAHNESTOCK, JOHN JAMES, F. J. KURTH, T. H. URDAHL.

One Year: L. N. HUNTER, C. O. MACKEY, R. D. MADISON, L. G. MILLER, L. P. SAUNDERS.

R. D. Madison

JOURNAL SECTION

(L. to R.) W. E. Zieber, T. H. Urdahl, M. K. Fahnestock, A. B. Algren, R. C. Cross, E. R. Queer, Miss I. M. Jahn, L. N. Hunter, C. M. Humphreys, L. G. Miller, R. D. Madison, W. O. Huebner, C. W. Johnson, John James

Technical Advisory Committees

Air Cleaning: R. S. Dill, *Chairman;* L. L. Dollinger, Jr., O. C. Eliason; R. S. Farr, C. D. Graham, A. B. Hubbard, R. E. Hunsaker, J. W. May, K. W. MacKenzie, R. H. Mills, G. W. Penney, C. B. Rowe, L. P. Saunders*, R. P. Warren, R. D. Wood.

Air Distribution: Ernest Szekely, *Chairman;* N. E. Berry, R. M. Conner, S. H. Downs, F. B. Holgate, F. J. Kurth*, J. N. Livermore, R. D. Madison*, G. E. McElroy, L. G. Miller*, D. W. Nelson, W. A. Pownall, G. B. Priester, T. H. Troller.

Air Sterilization: W. F. Wells, *Chairman;* W. T. Anderson, Jr., L. J. Buttolph, C. W. Flood, Jr., Leonard Greenburg, M.D., B. H. Jennings, E. R. Queer*, J. A. Reyniers, Mildred W. Wells, M.D., C.-E. A. Winslow.

Combustion: R. K. Thulman, *Chairman;* P. R. Achenbach, J. F. Barkley, L. L. Connell, R. C. Cross*, R. B. Engdahl, R. C. Johnson, W. B. Kirk, J. W. Miller, W. M. Myler, Jr., H. J. Rose, K. O. Schlentner, E. T. Selig, Jr., T. H. Smoot, E. C. Webb.

Cooling Load: W. E. Zieber*, *Chairman;* C. M. Ashley, R. D. Blum, W. F. Friend, R. H. Heilman, H. W. Heisterkamp, R. C. Jordan, C. F. Kayan, J. N. Livermore, R. H. Lock, C. O. Mackey*, J. P. Stewart.

Corrosion: L. F. Collins, *Chairman;* A. S. Gates, Jr., A. E. Griffin, E. W. Guernsey, R. L. Hallett, P. G. Kayser, H. A. Pray, F. N. Speller, T. H. Urdahl*.

Heat Flow Through Glass: R. A. Miller, *Chairman;* A. B. Algren*, A. H. Baker, F. L. Bishop, Jr., J. E. Frazier, J. S. Herbert, E. H. Hobbie, L. K. Jones, C. O. Mackey*, F. W. Preston, W. C. Randall, C. A. Richardson, Vic Sanders, H. B. Vincent, G. B. Watkins, F. C. Wood.

Heating Load: T. F. Rockwell, *Chairman;* C. M. Ashley, R. A. Biggs, A. I. Brown, R. C. Cross*, R. S. Dill, W. S. Harris, J. M. Hartman, L. G. Miller*, S. I. Rottmayer, M. W. Shears, G. H. Tuttle.

Heat Pump: R. C. Jordan, *Chairman;* E. R. Ambrose, R. D. Blum, C. F. Boester, C. W. Brown, W. F. Friend, M. C. Giannini, J. D. Kroeker, E. P. Palmetier, S. S. Sanford.

Industrial Ventilation: W. N. Witheridge, *Chairman;* A. D. Brandt, W. B. Harris, W. C. L. Hemeon, D. E. Humphrey, C. W. Johnson*, J. M. Kane, M. G. Kershaw, L. L. Lewis, L. G. Miller*, H. A. Mosher, B. F. Postman, B. R. Small, W. M. Wallace, II, R. P. Warren, H. E. Ziel.

* Member of Committee on Research.

Insulation: E. R. Queer*, *Chairman;* R. E. Backstrom, C. B. Bradley, R. B. Crepps, C. H. Jefferson, H. E. Lewis, L. E. Pasek, H. E. Robinson, Vic Sanders, T. D. Stafford, L. V. Teesdale, R. K. Thulman, T. T. Tucker, P. M. Woodworth.

Navy Studies: G. L. Tuve, *Chairman;* E. F. DuBois, M.D., J. D. Hardy, M.D., E. R. Queer*, L. E. Seeley, C. P. Yaglou.

Panel Heating and Cooling: G. D. Winans, *Chairman;* A. B. Algren*, R. L. Byers, P. B. Gordon, L. N. Hunter* *(Ex-Officio)*, John James*, J. S. Locke, John Magos, H. F. Randolph, Cyril Tasker *(Ex-Officio)*.

Group A—Heat Distribution Within and Behind the Panel: P. B. Gordon, *Chairman;* A. B. Algren*, R. A. Biggs, R. S. Dill, H. L. Flodin, F. E. Giesecke, W. S. Harris, C. F. Kayan, G. D. Lain, R. L. Maher, C. W. Meininger, D. L. Mills, C. W. Nessell, P. S. Park, S. I. Rottmayer, E. E. Scott, S. K. Smith, R. K. Thulman, S. M. Van Kirk, W. J. Widmer, G. L. Wiggs, L. H. Yeager.

Group B—Heat Transfer Between the Panel and the Space: John James*, *Chairman;* H. F. Randolph, *Vice Chairman;* C. M. Ashley, J. T. Bergen, C. F. Boester, Wharton Clay, Linn Helander, A. L. Jaros, Jr., H. A. Lockhart, C. O. Mackey*, R. A. Miller, G. W. Penney, G. G. Sward, J. M. van Nieukerken.

Group D—Controls: J. S. Locke, *Chairman;* A. B. Algren*, H. W. Alyea, S. D. Browne, R. L. Campbell, C. M. Garner, P. B. Gordon, W. J. Hajek, A. J. Keating, W. H. Kliever, H. T. Kucera, P. F. Neess, R. S. Penn, J. K. M. Pryke, E. J. Ritchie, C. W. Signor, N. D. Skinner, A. S. Widdowfield.

Physiological Research: C.-E. A. Winslow, *Chairman;* Thomas Bedford, M.D., Capt. A. R. Behnke, MC, USN, E. W. Brown, M.D., A. C. Burton, M.D., E. F. DuBois, M.D., A. P. Gagge, R. W. Keeton, M.D., D. H. K. Lee, M.D., L. G. Miller*, André Missenard, Charles Sheard.

Sorbents: John Everetts, Jr., *Chairman;* F. C. Dehler, M. C. Giannini*, C. H. B. Hotchkiss, E. R. McLaughlin, J. C. Patterson, G. L. Simpson, F. J. Swaney.

Sound Control: T. A. Walters, *Chairman;* C. M. Ashley, P. H. Geiger, F. B. Holgate, W. H. Hoppmann, II, G. C. Kerr, R. D. Madison*, J. S. Parkinson, T. H. Troller.

Weather Design Conditions: T. H. Urdahl*, *Chairman;* J. C. Albright, P. D. Bemis, H. S. Birkett, J. P. Fitzsimons, J. H. Hart, W. L. Holladay, W. M. Wallace, II.

INSTITUTIONS COOPERATING WITH THE COMMITTEE ON RESEARCH

Case Institute of Technology, Cleveland, O.: Air Distribution by Slots and Perforated Panels; Heat Transfer of Evaporating Refrigerants.

Columbia University, New York, N. Y.: Study, by Means of the Electric Analogger, of the Effect of Idle Tubes on the Heat Flow and Temperature Field in a Concrete Slab.

Cornell University, Ithaca, N. Y.: Effects of Size of the Sol-Air Thermometer on Measurements of Sol-Air Temperature.

Kansas State College, Manhattan, Kan.: Projection of Heated and Cooled Air Streams; Studies of the Radiant Effect in Sensory Reactions of Comfort.

University of Illinois, College of Medicine, Chicago, Ill.: Physiological Adjustments of Human Beings to Sudden Change in Environment.

Michigan State College, East Lansing, Mich.: Flow Losses in Branch Take-Off Fittings.

RESEARCH PAPERS—1948

1. Solar and Total Heat Gain Through Double Flat Glass, by George V. Parmelee and Warren W. Aubele (Cleveland, O.) (June 1948, JOURNAL SECTION).
2. Comparison of Physiological Adjustments of Human Beings During Summer and Winter, by Nathaniel Glickman, Tohru Inouye, Robert W. Keeton, M.D., (Chicago, Ill.) and Maurice K. Fahnestock (Urbana, Ill.) (August 1948, JOURNAL SECTION).
3. Physiologic Adjustments of Normal Subjects and Cardiac Patients to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, M.D., I. R. Callen, M.D., F. K. Hick, M.D., and M. K. Fahnestock. (February 1949, JOURNAL SECTION).
4. Overall Coefficients for Flat Glass Determined Under Natural Weather Conditions, by G. V. Parmelee and W. W. Aubele (Cleveland, O.) (December 1948, JOURNAL SECTION).
5. The Internal Performance of an Induced Draft, Packed Cooling Tower, by N. W. Snyder (Berkeley, Calif.) (February 1949, JOURNAL SECTION).
6. Heat Transfer Rates for Refrigerant Boiling in Horizontal Tube Evaporators, by L. G. Seigel, W. L. Bryan and M. C. Huppert (Cleveland, O.) (January 1949 JOURNAL SECTION).
7. Turbulence in Air Distribution, by H. B. Nottage (Cleveland, O.).

Society member is entitled to one free copy of any technical publication, including bulletins and progress reports, if he requests the same. Notice of the availability of these publications is announced to the membership through the JOURNAL.

It seems that the membership is not as well informed on the results of research as it should be, although all results are published in the form of technical papers. Consequently, steps are being taken by the Society to make available in a condensed and less technical form the contents of many of the technical papers for the benefit of those members who may have only a casual interest in a subject.

The members of the Committee on Research and the Technical Advisory Committees have proposed to the Program and Papers Committee several subjects and authors for papers for future Society meetings.

Future Program

The results of the recent survey of the Society membership regarding our research activities will be carefully analyzed. If this survey indicates there is an important difference of opinion between the thinking of the members of the Society and the Committee on Research or the Regulations as set forth in our Constitution, some action will have to be taken to reconcile the same. This may require a change in our policy or a determined effort to sell our members on our aims and objectives. It may prove the importance of our finding a way of informing our members more fully regarding what is going on.

On October 4, a budget of $191,550.00 for the ensuing year was submitted to Council and approved. This represents a substantial increase over last year's budget and requires increased support on the part of our members and of industry. Our enlarged research program places a greater responsibility on the members of the Committee on Research and the Technical Advisory Committees. Of the amount given, $48,700.00 is for research required by our Navy contract, and $142,850.00 is for other research.

Ways and means are being considered to make it possible for more of our Society members to participate in the planning of our research work. Many Society members who are not on Technical Advisory Committees have expressed an interest in the work of those Committees. Consequently, as an experiment at our Annual Meeting, members of the Society are being given an opportunity to attend the meetings of the Technical Advisory Committees. It is realized that this has both advantages and disadvantages, and if it should prove undesirable to continue this as a Society policy, it may be necessary to expand the membership of a number of the Technical Advisory Committees.

Laboratory Staff

The Committee on Research wishes to acknowledge the excellent cooperation of the Laboratory staff, under the able direction of Cyril Tasker, director of research. The interest shown by each staff member in his work, and the teamwork that has been demonstrated by the staff as a whole, have been very gratifying. The Committee recommended the reappointment of Mr. Tasker as Director of Research for the ensuing year, which action has been approved by Council.

An agreement has been worked out with the faculty of Case Institute of Technology whereby members of the staff of the Research Laboratory can undertake graduate studies at Case Institute. Some of the members of the staff are taking advantage of this opportunity.

The Report of the Director of Research, which follows, includes summaries of the work at the Laboratory and cooperating institutions, and of the Technical Advisory Committees.

Report of the Director of Research

by Cyril Tasker

The year 1948 has seen a significant expansion of Society research both at the Cleveland Laboratory and at a number of cooperating institutions. If one can use expenditures as an index of the activity of a research program, Society research activities in 1948 increased by about 25 percent over 1947. Actually much of this expansion took place in the second half of the year when a number of new projects were initiated and others accelerated as increased industry participation made it possible to add both staff and equipment.

Perhaps the two most important steps taken in 1948 were the signing of a contract with the Office of Naval Research to undertake work in the field of physiology and human calorimetry, and the continuous activity at the Laboratory on the study of heat distribution within and behind a concrete panel, which forms an important first step in the Society's overall program on panel heating and cooling. The studies on solar radiation transmission through glass building members continued throughout the year with additional staff engaged from June onwards. The program on air cleaning constituted a full-time activity in 1948 and showed signs of real progress as the year ended. We were able to initiate an experimental program on cooled ventilation jets during the summer and maintain full-time activity from October onward. An important investigation into the burning characteristics of fuel oil was undertaken under the sponsorship of the Oil Heat Institute.

Cooperative research was noticeably expanded in 1948 and at the year's end eight studies were in progress at six institutions. Physiological studies dealing with shock effect, under way at the College of Medicine, University of Illinois, were supported by a grant-in-aid from the U. S. Public Health Service effective as of July 1, 1948. Several phases of an overall program of research into air distribution were under attack at Case Institute of Technology, Kansas State College, and Michigan State College. The panel heating studies at the Laboratory were augmented by studies at Kansas State College and Columbia University. At Cornell University further studies were being made of the significance of various factors in solar heat gains. The U. S. Weather Bureau provided basic data to be used in a cooperative study at Tulane University of sol-air temperatures for a hot and humid climate. One phase of the study of the heat transfer coefficients of evaporating refrigerants was completed at Case Institute of Technology.

Staff and Facilities

Additions made to the engineering and technical staff during 1948 brought the full-time staff at the end of the year to twenty-three. Of these, eleven are graduate scientific staff. Additional part-time staff were used during the year as required.

Increased office facilities have been provided, improvements made to the laboratory facilities, and permanent equipment added as required, so that the laboratory inventory now stands at $20,600 after due allowances for depreciation.

Publications

Bulletin No. 1 *Heat Transmission Through Glass* came off the press at the close of the year and already a substantial number of copies have been distributed to Society members who requested both this bulletin and the two that preceded it: No. 2 *The Thermal Properties of Building Materials Used in Heat Flow Calculations* and No. 3 *Forced Convection Heat Transfer from Flat Surfaces.* Seven research papers were released from the Laboratory and

cooperating institutions, two were presented at the Semi-Annual Meeting and five prepared for presentation at the 55th Annual Meeting. Titles and authors are listed elsewhere.

The Guide

In addition to assistance in the review and revision of a number of Guide chapters by Laboratory staff members, a closer liaison has been developed between the Guide Publication Committee and the various Technical Advisory Committees. H. B. Nottage, as a member of a special sub-committee, undertook a study of existing psychrometric charts and the bases on which they were developed. This Committee has recommended that the Society undertake the development of a new psychrometric chart which would lend itself more readily to the graphical solution of air conditioning problems than the Goff diagram now included with the Guide. A start on this project will, it is hoped, be made during 1949.

Studies at the Laboratory and at Cooperating Institutions

Only the principal highlights of the projects under way at Cleveland, and at the universities cooperating with the Society in its research program, will be covered here. Progress reports on many projects have been distributed more widely this year than ever before, and under present Council regulations any Society member can secure progress reports as desired. Fuller details will be found in research bulletins and papers or may be secured by members on request.

Air Distribution

Ventilation Jets

As the first step in a long-term plan for practical research in room air distribution, equipment for the study of chilled air jets discharging into large spaces was constructed and assembled during the latter part of the year. Instrumentation of the test equipment is proceeding in order to enable measurement to be made of temperature, velocity and turbulence intensity in the jet region.

There is still much to be done on the problem of the measurement of low velocity air streams. The Society has been instrumental in the past in adding to the sum of knowledge in this field; the present activities are designed to investigate the significance of some of the important elements in this problem and attempt to solve them.

An extensive literature survey, covering about 150 references, undertaken by Mr. Nottage, is nearing completion. It has already proved of considerable value in defining the experimental program for the Laboratory. It is hoped that it will be possible to make this available in published form during 1949, so that it may be of lasting and timesaving value for those undertaking investigations into air distribution problems. This type of activity is one by which the Society can very properly serve the industry at large; it is seldom undertaken by industrial organizations, and even when it is carried on, the results seldom find their way into print.

Mr. Nottage has presented some views on turbulence in air distribution at the 55th Annual Meeting.

Slots

The studies on slots described at the 54th Annual Meeting were continued at Case Institute of Technology during the summer. Investigations were conducted on the air stream performance for a slot in the side of a tapered duct, the purpose being to obtain constant discharge angles along the slot in order to facilitate the application of turning vanes. Performance curves developed from theoretical considerations were checked by test results. This work is being prepared for presentation as a research paper.

Perforated Panels

Tests were also completed at Case Institute on eight perforated panels (thick transite and thin metal).

Residual velocity measurements were conducted in the constant-velocity zone near the panel, and in the zone located a short distance from the panel in which the residual velocity varies inversely with distance. The results for the residual velocities of the panels tested were found to plot on dimensionless parameters involving percentage free area, coefficient of discharge, panel area, velocity at the perforations, and distance from the panel.

Tests were conducted with air streams projected into an open space and into an enclosed space with the distance between floor and ceiling fixed. Velocity measurement was obtained with commercial-type heated thermocouple, heated thermometer, and bridled vane anemometers.

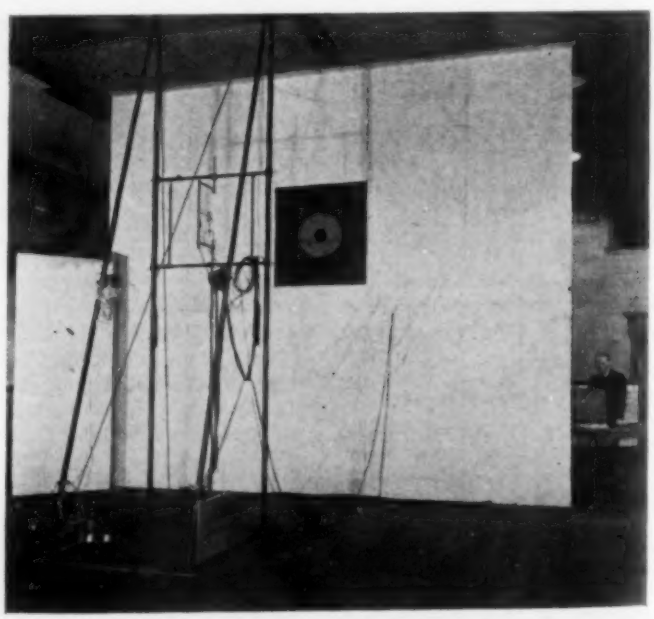
The stream profiles for perforated panels will be correlated at some future date with profiles of other type outlets in a report dealing only with stream envelopes.

Downward Projection of Heated Air

Studies reported upon at the 54th Annual Meeting continued during 1948 using the 5.75 in. convergent nozzle. Discrepancies were found between the values for entrainment obtained by direct measurement and the values calculated from heat balances. These discrepancies are being further investigated.

Tests were also made, both with room air and with heated air, using the convergent nozzle with the tubular extension of about one orifice diameter in length. Difficulties encountered in these studies show again how unsatisfactory the present situation is regarding the measurement of low velocity air streams. These difficulties show also the need for reliable methods that will ensure reproducibility.

Space is now available for the erection of less restricted test facilities in which larger orifice diameters and throws of 40 ft downward might be investigated.



Room air distribution research
Test apparatus with traversing carriage for the study of chilled air jets in large spaces

Friction in Fittings

A study of friction losses through branch take-offs in rectangular ducts is under way at Michigan State College under Dean L. G. Miller. The Laboratory has made progress on the compilation and analysis of existing data on losses of through-flow fittings. Experimental studies, unavoidably deferred throughout this year, should be under way early in 1949.

Heat Flow and Heat Transfer

Film Coefficients

Studies reported on in Research Bulletin No. 3 were extended during 1948 to cover the combined effects of natural convection and low velocity air movement over a range of 30 to 300 fpm on heat transfer from a smooth surface. The results are now being analyzed preparatory to publication.

Solar Radiation Transmission

Even though weather conditions during 1948 were not quite as favorable for the studies on solar radiation transmission as they were in 1947, they constituted a full-time activity for three staff members. Two papers were prepared and Bulletin No. 1 released in published form.

Additional studies made on flat glass, in single sheets and in double glass combinations, covered transmittance for direct and diffuse solar radiation as well as total heat gain. Overall coefficients of heat transfer for single and double glass were also determined.

The investigation of solar heat transmission through glass block which commenced in May has required a considerably more detailed study than was necessary with flat glass. Four different patterns of 8 in. block were studied during the latter half of the year; the investigation is continuing as weather permits; preliminary data will be available in published form by the summer of 1949.

To measure incoming radiation from all sources on the outdoor surface of the panels under test, a convection-compensated radiometer has been designed and constructed at the Laboratory and is now undergoing calibration. It is hoped by this instrument to establish the convective and radiative heat exchanges at the outer surface of the glass as accurately as they are now established for the heat exchanges between the indoor surface of a test panel and the calorimeter used in these studies.

Securing this information is essential to a better understanding of all the factors that enter into questions of heat gains or losses from sunlit glass areas. It plays an important part in the final objective of developing design data.

Industry interest in, and support for, these studies has been well maintained during the year and has greatly encouraged the staff engaged in this work.

Cooling Loads—Periodic Heat Flow

At Cornell University, Prof. C. O. Mackey is studying the effect of the size of the sol-air thermometer, which was developed by him partially under a cooperative research agreement with the Society, on the sol-air temperatures obtained under natural weather conditions. The U. S. Weather Bureau has compiled the basic data for the determination of sol-air temperatures for New Orleans. It is expected that these will be computed and made available by the summer of 1949 and thus provide these data for a hot and humid area.

It would appear as if we had reached the stage where careful experimental studies should be arranged to supplement the information developed by analytical treatment. Periodic heat flow is still but imperfectly understood and appreciated by many air conditioning engineers. Yet it is of real significance in the development of cooling load calculations.

Insulation

The studies mentioned in the 1947 report were completed during the year and plans are being made to present the results in a technical paper.

Heat Transfer Coefficients of Evaporating Refrigerants

Prof. Seigel and his colleagues of Case Institute presented at the 55th Annual Meeting a paper covering the results obtained to date in studies designed to investigate the effects on heat transfer of such factors as: loading; oil circulation with the Freon; ratio of heated length of tube to inside diameter; refrigerant flow; and temperature difference, tube to refrigerant.

Physiological Research

The studies being carried on at the College of Medicine, University of Illinois, under Dr. R. W. Keeton, received additional support during the latter half of 1948 through a grant-in-aid from the National Institute of Health, U. S. Public Health Service. Two papers were prepared: one was presented at the 1948 Semi-Annual Meeting; the second at the 55th Annual Meeting.

The first paper gave comparisons of physiological adjustments of human beings to sudden changes in environment during summer and winter; the second discusses adjustments of young normal subjects and of cardiac patients (aged 25 to 72 years) to such changes. It is of interest to learn that cardiac patients reacted in the same manner as young normal subjects to sudden changes in environmental temperature, and that no ill effects are apparent as a result of entering and leaving air conditioned spaces during the hot summer months. The studies are continuing with both healthy and physically impaired subjects of both sexes, using both special research clothing and normal summer weight clothing.

Human Calorimetry (O.N.R. Contract)

On June 20, a contract was signed with the Office of Naval Research to undertake at the Cleveland Laboratory a basic investigation in physiology. The main purposes of this investigation were:

(a) To develop a calorimeter for a continuous registration of human heat exchange and its components.

(b) To study human heat exchange and its factors under a variety of experimental conditions including extreme and sudden environmental stresses.

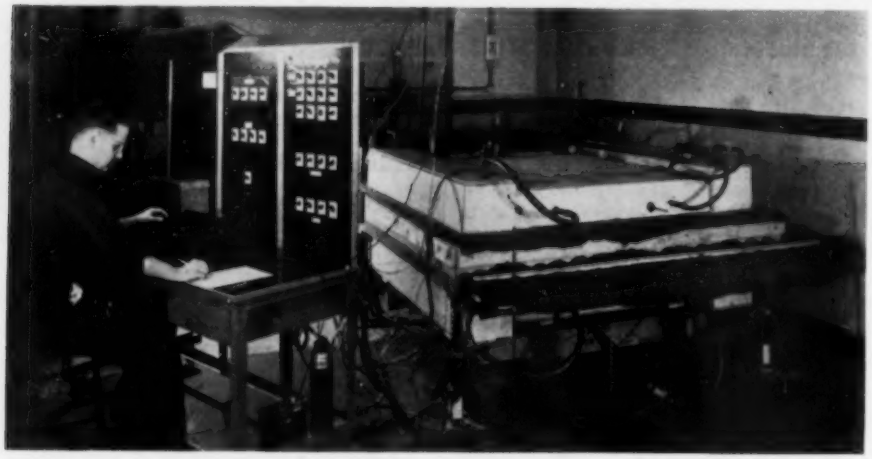
R. G. Huebscher of the Laboratory staff was assigned to these studies, and Dr. T. H. Benzinger, of the staff of the Naval Medical Research Institute, Bethesda, is spending part of his time in Cleveland working with the Laboratory staff. The basic principle of this calorimeter was developed by Dr. Benzinger, originally at the University of Heidelberg, Germany, and later at Bethesda.

There are a great number of interesting and important engineering problems as well as problems of instrumentation involved in translating the basic principles of such a device into a full-sized calorimeter capable of holding a subject and operating to the accuracy and within the limits set up in the project outline. A special Advisory Committee was appointed under the chairmanship of Prof. G. L. Tuve, and at the end of the year sufficient progress had been made to enable the Laboratory to submit a tentative design for the full-sized calorimeter to this Committee for study and approval.

All operating costs on this project are being covered by funds made available through the Office of Naval Research.

Panel Heating and Cooling

The broad outlines of the Society's research program in panel heating were set out in the Annual Report of the Committee on Research for 1947. There have been no serious changes in these objectives since the report was presented, and the first part of the program, under Group A, has been under way continuously ever since. The first concrete slab was poured in March, and as soon as it was dried out and the instrumentation completed, tests were begun. So far as the Laboratory staff know, the present experimental techniques, though by no means new, have not previously been applied to a problem such as this, namely the measurement of the heat flow within, behind and from a 6 ft x 6 ft concrete slab



Panel heating research

Slab No. 1 under test with top side insulated and upper and lower absorbers in use

weighing over 4200 lb. Some instrumentation and control problems were therefore to be expected. They were met and overcome, and at the end of the year tests had been completed on this first slab at 4 in., 8 in., 12 in. and 16 in. tube centers, over a wide range of tube temperatures and with varying heat outputs. Tests of the effects on the heat flow of various thicknesses of insulation are currently under way. A study of floor coverings will follow. Exhaustive tests made on the first slab when operating with 4 in. tube centers gave data which made it possible to reduce the number of tests at other tube centers and under other conditions.

A second slab, much thinner, has been poured. All thermocouples have been inserted in the interior and on the surface, and the slab is now being dried out ready for testing.

Analytical studies have been carried on to parallel the experimental investigation, but so far no conclusive results have been obtained.

At Columbia University, Prof. Carl F. Kayan is investigating, by electrical analogy methods, the effect of idle tubes on the heat flow in a slab.

An extensive bibliography of articles and references on panel heating and cooling up to the end of 1947 was prepared by the Laboratory staff and made available in mimeographed form to Society members and the industry in general. A large number of copies have already been distributed. It is planned to bring this bibliography up to date yearly.

The program under way at Kansas State College consists of an analytical study of the psychrometric room to determine the radiation interchange between an object within a bare room and its environment, and a test program within the room to study the various means of measuring surface temperatures and the radiating properties of a room. Prof. Tripp is now revising a report covering *Radiation Shape Factors Between a Plane Wall and a Globe Thermometer*, which he submitted for review during the year. Tests with limited objectives are under way in the psychrometric room; results will be reported later.

During the year a Danish engineer, Vagn Korsgaard, served for a few months on the staff of the Laboratory. During that time he prepared, with the assistance of other members of the Laboratory staff, three papers entitled:

(a) The Necessity of Using a Directional Mean Radiant Temperature to Describe the Thermal Conditions in Rooms

(b) A New Radiometer Measuring Directional Mean Radiant Temperatures

(c) Resultant Temperature and Its Measurement—A Summary of Two Danish Papers.

These will appear in the JOURNAL in the near future.

Air Cleaning

Steady progress has been made during 1948 in the program designed to develop sound and acceptable codes for the testing and rating of air cleaning devices. Changes were made to the test equipment described and illustrated in the 1947 Annual Report, and techniques developed during the year for both velocity and dust distribution measurements were checked under a variety of operating conditions.

It would be of considerable assistance in outlining some of the further steps to be taken in this investigation and defining the degree of accuracy to be sought in filter testing if an answer could be obtained to the question, *Can the industry tell us what it wants a filter to do?* This answer should come from filter users, perhaps more than from filter manufacturers, though it is doubtful if most users know what degree of cleaning they require and can afford to demand.

Certain filter manufacturers and some large users have continued to take a great interest in and to give active support to this program during the year. Clean air is not only important to many industries but is probably of significance in questions of health. Wider participation in this program would enable a more concentrated attack to be made on the many angles involved.

Oil-Heat Institute

Work on the Combustion Reference Test Unit mentioned in the 1947 report continued during the spring of 1948. The unit was transferred to Cleveland in April and set up with improved instrumentation. Tests made with three widely different oils showed that the unit had good possibilities of evaluating the burning characteristics of fuel oils suitable for domestic use. Additional instrumentation seems desirable to insure reproducibility of test results, especially in the matter of smoke density measurement. Certain instrument makers are cooperating with the Institute and the Society to this end. Further work has been approved by the Engineering Committee of the *OHI* and will be under way early in 1949.

General

Director's Visit to England

The writer left the United States in mid-August and returned in mid-October after visiting a number of research stations, laboratories, colleges, and other institutions in England where studies of interest to the Society were in progress. He discussed the subject of research for the heating, ventilating and air conditioning engineer before a special meeting of the *Institution of Heating and Ventilating Engineers* on September 29 in London dealing in some detail with the Society's research program. He also spoke before the Fuel Luncheon Club on October 5 and the Rumford Club on September 23, both in London.

Liaison was established with most, if not all, of the principal investigators in the Society's field of interest, and mutual exchange of reports and memoranda arranged. This has already begun and reports, received well in advance of general publication, are proving of interest in the Society's research program.

A report *British Research in Heating and Ventilating in 1948* will be available in mimeographed form early in 1949. Copies may be obtained from the Laboratory.

Staff members have addressed five chapters during 1948 on various phases

of the research program. Contact has been established and/or maintained with the Department of the Army and with the Navy Department.

Acknowledgments

The expansion of Society research during 1948 was only possible because the Laboratory staff received every encouragement and support from the Society officers and Council and especially from the chairman and members of the Committee on Research and the members of the various technical advisory committees. On behalf of the staff the writer expresses deep appreciation of this confidence. Research accounting was transferred to Cleveland as of July 1, thus enabling a closer check to be kept on project expenditures and general operating expenses. The writer gladly takes this opportunity to express his thanks to all Laboratory staff members for their loyal services and especially to Mr. Humphreys who acted as Director during his absence from the country.

PARTICIPANTS IN THE 1948 ASHVE RESEARCH PROGRAM
November 1, 1947—October 31, 1948

Aerofin Corp.
Air Filter Institute
Air-Maze Corp.
Detroit Air Filter Co.
Farr Co.
Owens-Corning Fiberglas Corp.
Research Products Corp.
Airkem, Inc.
*American Air Filter Co., Inc.
*American Blower Corp.
*The American Brass Co.
**American Radiator & Standard Sanitary Corp.
*American Structural Products Co.
*American 3 Way-Luxfer Prism Co.
*American Window Glass Co.
*Anemostat Corp. of America
Association of American Railroads
Automatic Burner Corp.
Barber-Colman Co.
Barnes & Jones, Inc.
Bastian-Morley Co., Inc.
Bayley Blower Co.
*Bell and Gossett Co.
*Bell Telephone Laboratories, Inc.
Bethlehem Steel Co.
*Blue Ridge Glass Corp.
Buffalo Forge Co.
Carnegie-Illinois Steel Corp.
**Carrier Corp.
Chicago Pump Co.
Columbus Heating & Ventilating Co.
Copper & Brass Research Association
Ampco Metal, Inc.
Bohn Aluminum & Brass Corp.
Bridgeport Brass Co.
The Bridgeport Rolling Mills Co.
The Bristol Brass Corp.
Chase Brass & Copper Co., Inc.
Chicago Extruded Metals Co.
Detroit Gasket & Manufacturing Co.—Extruded Metals Div.
The Drawn Metal Specialty Co., Inc.
Wilbur B. Driver Co.
Driver-Harris Co.
The Electric Materials Co.
Harvey Machine Co., Inc.
Hudson Wire Co.
C. G. Hussey & Co.
Ilsco Copper Tube & Products, Inc.
Lewin-Mathes Co.
The Linderme Tube Co.
The Mackenzie Walton Co., Inc.
The Miller Co.
Mueller Brass Co.
The National Copper & Smelting Co.
New England Brass Co.
The New Haven Copper Co.
Penn Brass & Copper Co.
Phelps Dodge Copper Products Corp.
The Phosphor Bronze Corp.
The Plume & Atwood Mfg. Co.
Reading Tube Corp.
Revere Copper and Brass Inc.
The Riverside Metal Co.
Roberts Tube Works
Scovill Manufacturing Co.
The Seymour Manufacturing Co.
Small Tube Products, Inc.
Somers Brass Co.
Stamford Rolling Mills Co.
The Thinsheet Metals Co.
Titan Metal Manufacturing Co.
United Wire & Supply Corp.
Volco Brass & Copper Co.
Waterbury Rolling Mills, Inc.
A. H. Wells & Co., Inc.
Western Brass Mills
Wolverine Tube Div.
Crane Co.
*Detroit Edison Co.
*Detroit Lubricator Co.
Detroit Stamping Co.
Duquesne Light Co.
Eastwood, Prof. E. O.
*Farr Co.
Fitz, Jean C.
Forslund Pump & Machinery Corp.
Frick Co.
*Frigidaire Div. of General Motors Corp.
G & O Manufacturing Co.
Heating, Piping & Air Conditioning Contractors National Association
Advance Heating & Air Conditioning Corp.
John F. Ahern Co.
Almirall & Co.
W. M. Anderson Co.
J. P. Baldwin Co.
W. E. Beggs Co.
Bon Heating Co.
Butters-Fetting Co.
Child & Scott Co., Inc.
Commonwealth Piping & Ventilating Co.
T. J. Conner, Inc.
Courter & Co.
E. N. Cunningham Co.
M. J. Daly & Sons Co.
Eastern Mechanical Corp.
A. C. Eynon Plumbing Co.
Gregory & Graham, Inc.
Hanson & Gavin
Hayes, Joseph G.
Heating, Piping & Air Conditioning Contractors Chicago Association
Heating, Piping & Air Conditioning Contractors Cincinnati Association
Heating, Piping & Air Conditioning Contractors Detroit Association
Heating, Piping & Air Conditioning Contractors Minneapolis Association
Heating, Piping & Air Conditioning Contractors Philadelphia Association
Heating, Piping & Air Conditioning Contractors Pittsburgh Association
Heating, Piping & Air Conditioning Contractors Association of Southern California
Hosler & Pearson, Inc.
Hudson, Frank
Jarvis Engineering Co.
Johnston Heating Co.
Jung, John S.
Knab, E. A.
Landers Plumbing & Heating Co.
Lawton, Frank C.
Harry Lee Plumbing & Heating
Lind, Elmer C.
Maag, E. J.
Mimmack Heating Co.
Monihan-Stauffacher Co.
Morris & Eagan Co.
Mueller, H. E.
H. G. Newman Plumbing Co., Inc.
H. F. Newstead Heating Co.
J. J. Nolan & Co.
Philadelphia Piping & Equipment Co.
William A. Pope Co.
Power Piping Corp.
Raisler Corp.
George F. Reeke Co.
C. R. Reichel Engineering Co.
Reliance Heating & Air Conditioning Co.
Wray M. Scott Co.
Sheridan Heating & Plumbing Co.
W. H. Simpson, Inc.
Rowland Tompkins & Son, Inc.
Werner, Paul E.
Fred Williams, Inc.
Wolff & Munier, Inc.
The Heil Co.
Ilg Electric Ventilating Co.
Illinois Testing Laboratories, Inc.
Inland Steel Co.
The Institute of Boiler & Radiator Manufacturers
Burnham Corp. (Boiler Div.)
Crane Co.
Dunkirk Radiator Corp.
The Eastern Foundry Co.
The Gurney Foundry Co., Ltd.
International Heater Co.
Kohler Co.
The National Radiator Co.
Frank Prox Co.
The H. B. Smith Co.
Spencer Heater Division—Avco Mfg. Co.
Taylor-Forbes, Ltd.
Thatcher Furnace Co.
U. S. Radiator Corp.
Utica Radiator Corp.
Weil-McLain Co.
*Johnson Service Co.
Joy Manufacturing Co.
Keeney Publishing Co.
*Kewanee Boiler Corp.
Kinetic Chemicals, Inc.
Kramer Trenton Co.
*Libbey-Owens-Ford Glass Co.
The Marley Co., Inc.
McQuay, Inc.
Mellish and Murray Co.
Metal Window Institute
*Minneapolis-Honeywell Regulator Co.
*Mississippi Glass Co.
Nash Engineering Co.
National Mineral Wool Association
John J. Nesbitt, Inc.
Oil-Heat Institute of America, Inc.
*Owens-Corning Fiberglas Corp.
*Pennsylvania Wire Glass Co.
*Perfex Corp.
*Pittsburgh Corning Corp.
*Pittsburgh Plate Glass Co.
Powers Regulator Co.
*The Pyle-National Co.
Ranco, Inc.
*Research Products Corp.
Rhoton, W. R.
Richmond Radiator Co.
Sears, Roebuck and Co.
Spencer Thermostat Co.
Surface Combustion Corp.
*H. A. Thrush and Co.
Timken-Detroit Axle Co.
The Trane Co.
The Tremco Manufacturing Co.
*U. S. Navy—Office of Naval Research
*U. S. Public Health Service—National Institute of Health
Viking Air Conditioning Corp.
*Westinghouse Electric Corp.—Sturtevant Div.
White-Rodgers Electric Co.
L. J. Wing Manufacturing Co.
*Young Radiator Co.

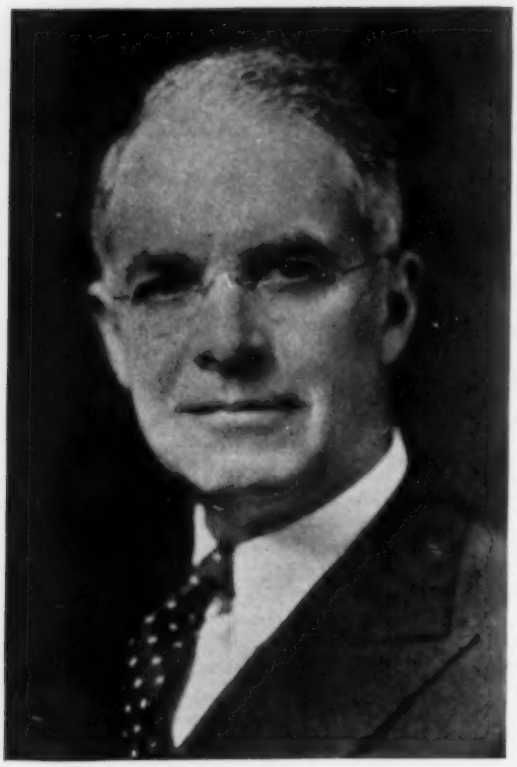
*Earmarked participation.
**General and Earmarked participation.

PARTICIPANTS IN THE 1949 ASHVE RESEARCH PROGRAM
November 1, 1948—December 31, 1948

*American Air Filter Co., Inc.
American Iron & Steel Institute—
Committee on Steel Pipe Research
Bethlehem Steel Co.
Fretz-Moon Tube Co., Inc.
Jones & Laughlin Steel Corp.
Laclede Steel Co.
Mercer Tube & Manufacturing Co.
National Tube Co.
Pittsburgh Tube Co.
Republic Steel Corp.
Sharon Tube Co.
Spang Chalfant Division of the National Supply Co.
Wheatland Tube Co.
Wheeling Steel Corp.
Youngstown Sheet & Tube Co.
**Carrier Corp.
Chrysler Corp.—Airtemp Div.
*Farr Co.
*Fitzgibbons Boiler Co., Inc.
Heating, Piping & Air Conditioning Contractors National Association
City Plumbing & Heating Co.
Heating, Piping & Air Conditioning Contractors Memphis Association
Heating, Piping & Air Conditioning Contractors Milwaukee Association
Reliance Heating & Air Conditioning Co.
Spohn Heating & Ventilating Co.
C. P. Wright & Co.
*Owens-Corning Fiberglas Corp.
*Raytheon Manufacturing Co.
Servel, Inc.
*U. S. Navy—Office of Naval Research
*U. S. Testing Co.

OFFICERS FOR 1949

A · S · H · V · E



Alfred E. Stacey, Jr.
Syracuse, N. Y.
President

Lester T. Avery
Cleveland, Ohio
First Vice President

Lauren E. Seeley
Durham, N. H.
Second Vice President

Ernest Szekely
Milwaukee, Wis.
Treasurer

55th Annual Meeting Largest in Society History

Proceedings

A RECORD registration of 2508 members, ladies and guests insured excellent attendance at the technical sessions and all other events of the Society's 55th Annual Meeting held in the Stevens Hotel, Chicago, from January 24-27, 1949.

This became evident as every chapter was represented, from Ontario to Texas, and from New York to California.

Technical sessions, two luncheons, and the ladies' program gave everyone an educational and entertaining schedule. There were events at which members could be informed about research being carried on during the past year. There were other events at which members and guests could relax and meet old friends and make many new ones.

The climax of the Meeting was the annual banquet on Wednesday night, attended by 1100 persons in the grand ballroom of the hotel. Attendance at the 9th International Heating and Ventilating Exposition, which was held from January 24-28, was also a must on everyone's list of activities.

At the close of the four-day gathering, the new president, A. E. Stacey, Jr., Syracuse, N. Y., was hailed by the Society, and the retiring president, G. L. Tuve, Cleveland, was given a round of standing applause. The other officers installed were: Lester T. Avery, Cleveland, 1st vice president; L. E. Seeley, Durham, N. H., 2nd vice president; and Ernest Szekely, Milwaukee, treasurer.

Monday Events

The first assembly on Monday was a combined business and technical session, presided over by Prof. G. L. Tuve. An address of welcome was made by W. A. Kuechenberg, general chairman of the Illinois Chapter Committee on Arrangements. Mr. Kuechenberg officially welcomed members and guests in the name of the Illinois Chapter, whose president, C. M. Burnam, Jr., was introduced to members and guests.

Following reports by R. A. Sherman, chairman of the Finance Committee, and by A. V. Hutchinson, secretary of the Society, a report on research was read by L. N. Hunter, chairman of the Committee on Research (see p. 115).

Report of the President

It is one of the privileges of the President to learn many things about our Society that do not come to the attention of each individual member. Budgets and financial statements, reports of 50 or more committees, voluminous correspondence that contains a great variety of suggestions, comments and complaints; meetings, conferences and visits to Chapters—these are some of the ways the President gets his information.

One of the greatest problems of all your officers, committee chairmen and executive employees, is to keep the individual member of the ASHVE informed about Society activities. Of course the technical activities are most important, and these are well covered by technical papers, THE GUIDE, the TRANSACTIONS and the Research Bulletins. But we are proud that the ASHVE is a most democratic organization, and our members want to know what goes on—how the business of the Society is being managed, and whether its many problems are receiving due attention.

Keeping Members Informed

Now it is not easy to get the members to listen—and even harder to get them to read. All of us have developed a hard shell of self-protection against the millions of words of talk and print that come our way. When officers and committees get discouraged in their efforts to tell their story, a few members raise their voices to claim that things are being kept *from* them. Please be assured such isn't the case. It's just difficult to get the word around to all those that ought to be reached.

The present officers of your Society decided very early in the game that one of their most important jobs was to make it easy for the ordinary member to learn *all about* the workings of his Society. The Finance Committee started on the budget and on the financial statements, to reduce them to engineering language. A *research page* appeared in the JOURNAL, and Research Bulletins and reprints were offered to members for the asking. The President has sent out several general letters to all members, and the Membership Committee prepared a new booklet—under the title *The Aims and Activities of ASHVE*. A Committee on public relations was appointed, and that committee was reminded that the most important part of its public was our 6500 members. The fact that meetings of Technical Advisory Committees were open to interested members has been publicized, and the Chapter Relations Committee has furnished information about Society affairs to the dozens of speakers it sends to Chapters. Gentlemen, we have been trying to keep you posted, but we know there is much more to be done.

ASHVE Grows

It might be in order therefore, to mention to you at this time, several items in the year's accomplishments. As you know, practically all decisions and actions are made by Committees,

hence, I am only giving you a few *bits* of information, and you can easily get the complete stories from our many Committee reports.

We have had almost 1500 membership applications in the past year. Over half of our entire membership has joined us since the war. This growth has posed a lot of problems besides that of communication. In fact, I doubt if we can even maintain our identity when we're growing so fast. A very large percentage of our members have never served on a committee—any committee—even in a chapter. We now have more chapters than the officers can visit. Our headquarters office needs more room to handle the larger volume. Our reserve funds are no longer ample for our size.

Mortgage Paid

But, we are making progress, and we have some big accomplishments to our credit. In December, the mortgage on our Research Laboratory was paid off, and now our research activities do actually have a home of their own. Almost a hundred committee meetings were held in the Cleveland Conference rooms during the year, with members in attendance from all over the country and with a program that has hearty and *substantial* support from industry.

Publication Increased

We have been able to increase our publications program in ASHVE, while many others have been curtailing theirs because of higher costs. Our chapters and our members are becoming more conscious of their professional responsibilities in their several communities, and we're learning to cooperate with other groups of engineers, with doctors and public health authorities, with architects and owners, and with the Army and the Navy.

Broadened Research

Our research program has been broadened and our Technical Advisory Committees are exerting more influence, both on the Research Program and on THE GUIDE. We have a back-log of technical papers and THE GUIDE is one of the world's *best sellers* among technical books.

But the remarkable thing about our Society is the great enthusiasm and energy of its members. Dozens of committees are at work and the chapter officers, the Council, the headquarters staff and the research organization all work so hard and so fast that the President can't *possibly keep up* with all the happenings. But these three years of office have been both exciting and satisfying ones for me. The associations and friendships we all make as active workers in ASHVE are our most valued treasures. I wish I could call by name the individuals and groups that have done so much to make this past year's experience for me and for the Society a most gratifying one. I can only say a simple and sincere *thank you*. Individual officers can do little toward the success of our Society; it is the joint efforts of hundreds of enthusiastic workers, and the efficient functioning of our two excellent staff organizations that make possible our technical accomplishments on the one hand—but more important—our friendly Society.

Respectfully submitted

G. L. TUVE,
PRESIDENT

Report of Finance Committee

R. A. Sherman, chairman of the Finance Committee, reporting on the year's activities, described the committee's efforts to revise the details of the handling of funds and of reporting the financial transactions in order to make them more easily understood.

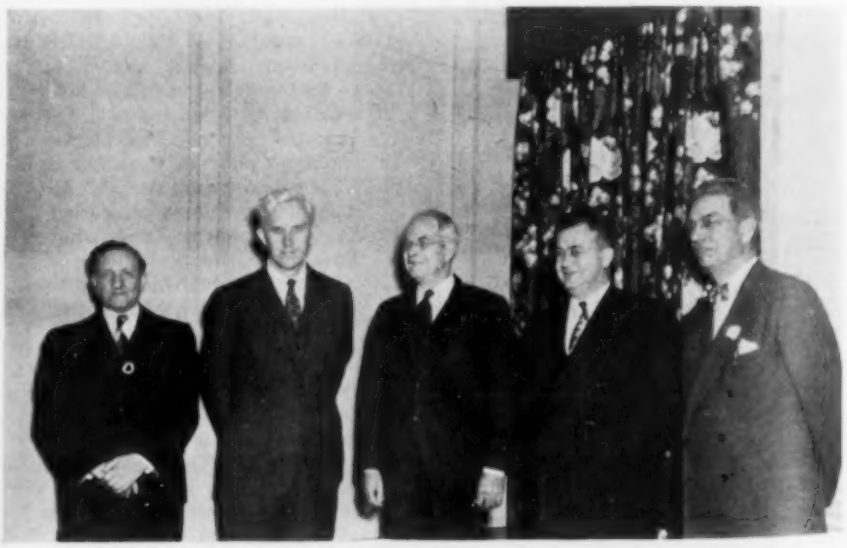
Mr. Sherman presented financial data in graphic form showing in percentages the sources of Society income. The calculations on a per member basis are made on 6,704 members of all classes as of October 31, 1948, Mr. Sherman said.

Cash receipts exceeded expenditures by $1,693.38, as compared to a budgeted prediction, when available reserves were considered, of $756.15.

In accordance with the action of the Council at the October 4, 1948 meeting, a balance of $7,000 payable on the mortgage of the laboratory, was paid on December 15.

Another important step made during the year 1948 was the return to the previous annual dues rate of $25.00 for members and associates. It is probably too early to know the full effect of this on the Society, Mr. Sherman stated. He pointed out that though it would probably mean a number of resignations, a large membership does not necessarily mean a strong Society either professionally or financially.

Complete details of the Society's income and expenditures are given in the report of the Certified public accountant which follows:



President Tuve with the Newly Elected Officers. ***(L. to R.)*** **Ernest Szekely (treasurer), Milwaukee; Prof. G. L. Tuve, Cleveland; A. E. Stacey, Jr. (president), Syracuse, N. Y.; L. E. Seeley (2nd vice president), Durham, N. H.; and Lester T. Avery (1st vice president), Cleveland**

Accountant's Report

Frank G. Tusa & Co.
Certified Public Accountants
52 William Street
New York

American Society of Heating and Ventilating Engineers
51 Madison Avenue
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of the American Society of Heating and Ventilating Engineers—New York, N. Y. and the related funds for the fiscal year ended October 31, 1948 and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1948. Also, for the fiscal year then ended, the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected and compared with the record of cash disbursements; the disbursements were supported by payment vouchers and the dues income from members and the interest income from investments was accounted for.

A Balance Sheet reflecting the financial condition of the Society as of the close of business October 31, 1948 is submitted herewith and your attention is directed to the following comments thereon:

CASH

The cash on deposit was verified by direct communication with commercial and savings banks and reconcilement of the balances reported to us with those reflected by the books of the Society. A schedule of cash is included as part of this report.

Checks representing the cash on hand for deposit were inspected by us and the petty cash counted.

MARKETABLE SECURITIES

The securities shown on the subjoined schedule were verified by direct communication with the Bankers Trust Co., where same are deposited for safe-keeping. This asset has been included in the balance sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

ACCOUNTS RECEIVABLE

A trial balance of the membership dues receivable taken as of the close of business October 31, 1948 was classified as to membership and aged as follows:

MEMBERSHIP	TOTAL
Members	$1,832.50
Associates	3,845.42
Juniors	357.50
Students	72.00
TOTAL	$6,107.42

AGING	
1948	$5,469.92
1947	588.00
1946	36.00
1945	13.50
TOTAL	$6,107.42

The reserves for dues and sundry accounts receivable doubtful of collection as provided in our opinion are ample.

INVENTORIES

The emblems and Transactions on hand on October 31, 1948 were counted by us. The Guide paper was verified by direct communication with the printers. All inventories were priced and computed by us. A schedule of Transactions inventoried follows:

Volume	Year	Quantity	Price	Amount
1-47	1895-1941	3,229	$ .40	$1,291.60
48	1942	100	1.42	142.00
49	1943	240	1.39	333.60
50	1944	197	1.82	358.54
51	1945	359	1.91	685.69
52	1946	392	1.67	654.64
		4,517		$3,466.07

DEPOSIT RECEIVABLE

The deposit placed with the United Air Lines in the sum of $425.00 was verified by direct communication.

ADVANCES

We have classified under exchanges advances made for the accommodation of council members reimbursed subsequently to the date of audit.

As at October 31, 1948 the participating employees to the retirement plan were indebted in the sum of $1,433.46 for contributions advanced by the Society to the Pension Fund. This sum is to be reimbursed to the Society by periodical deductions from wages of employees covered by the plan.

DEFERRED CHARGES

One-sixth of all subscriptions paid during the current fiscal period to H.P.A.C. were deferred as a future expense since the subscriptions are paid on a calendar year basis and the fiscal year of the Society ended on October 31, 1948.

TAXES

The sum of $524.20 represents Federal Income Taxes withheld from salaries paid to employees during the month of October 1948. The Society may be subject to possible Social Security tax assessments pending decision from the U.S. Treasury Department.

DUE RESEARCH

The balance due to the Research Laboratory as at October 31, 1948 is made up as follows:

40% of Dues Receivable from members and Associates		$2,162.45
Less: Depreciation charge for the current fiscal period	$1,446.42	
40% of dues prepaid by members and associates	393.24	1,839.66
BALANCE		$ 322.79

ACCRUED ACCOUNTS

The salaries and commissions have been computed and accrued in accordance with Finance Committee Report of April 8, 1948. The accrued expenses represent professional fees applicable to the current fiscal period.

DEFERRED INCOME

The prepaid dues and initiation fees by members and candidates for membership have been deferred to future operations. The membership classification of the dues prepaid by elected members follow:

MEMBERSHIP	AMOUNT
Members	$ 551.51
Associates	431.59
Juniors	151.00
Students	45.00
TOTAL	$1,179.10

MORTGAGE PAYABLE

The mortgage payable on the Cleveland real estate has been verified by direct communication with the Cleveland Trust Co. On December 15, 1948 this mortgage was completely satisfied.

RESERVE FOR TRANSACTIONS

We have reserved the sum of $24,000.00 to cover the publication of Transactions, Vols. 53 and 54 which are scheduled for publication in 1949. This sum includes an increase of $6,000.00 in excess of the 1947 Budget Provision to cover the estimated cost of publishing Vol. 53.

FUNDS

An analysis of the following Fund Accounts covering the changes that occurred therein during the fiscal year ended October 31, 1948 is attached hereto:

General Fund
Reserve Fund
F. Paul Anderson Fund
Property Fund
Mortgage Reduction Fund

In accordance with instructions received from the Finance Committee we have set up a Property Fund account under which we have classified the Land, Buildings, Furniture, Fixtures and Library.

Respectfully submitted.

FRANK G. TUSA & CO.
Certified Public Accountants

JOURNAL SECTION

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—NEW YORK, N. Y., October 31, 1948

ASSETS

GENERAL FUND				
CASH				
On Deposit		$27,101.47		
On Hand for Deposit		8,938.31		
On Hand		100.00	$36,139.78	
SECURITIES (AT COST)				
Securities (Market Value $37,113.65)		34,439.50		
ADD: Accumulated Interest	$3,240.90			
Accrued Interest	125.00	3,365.90	37,805.40	
ACCOUNTS RECEIVABLE				
Membership Dues	6,107.42			
LESS: Reserve for Doubtful	1,521.68	4,585.74		
Advertisers and Sundry Debtors	9,671.16			
LESS: Reserve for Doubtful	1,000.00	8,671.16	13,256.90	
INVENTORIES				
Transactions and Paper		3,744.71		
Emblems		328.55		
Guide Paper		1,140.00	5,213.26	
DEPOSIT RECEIVABLE			425.00	
EXCHANGES			343.85	
ADVANCES TO PENSION FUND FOR EMPLOYEES			1,433.46	
DEFERRED CHARGES				
Prepaid H.P.A.C. Subscriptions			2,179.73	$ 96,797.38
PROPERTY FUND				
Land and Buildings		84,920.96		
LESS: Reserve for Depreciation		3,908.07	81,012.89	
Furniture and Fixtures		3,385.01		
LESS: Reserve for Depreciation		1,258.73	2,126.28	
Library			300.00	83,439.17
RESERVE FUND				
Cash on Deposit			5,134.26	
Securities at Cost (Market Value $44,865.74)		40,145.00		
ADD: Accumulated Interest		4,720.74	44,865.74	50,000.00
F. PAUL ANDERSON FUND				
Cash on Deposit		60.00		
Cash on Hand for Deposit		25.00	85.00	
Securities at Cost (Market Value $947.00)		1,000.00		
ADD: Accrued Interest		12.50	1,012.50	1,097.50
MORTGAGE REDUCTION FUND				
Cash on Deposit				162.78
				$231,496.83

LIABILITIES AND FUNDS

LIABILITIES				
FEDERAL WITHHOLDING TAX			$524.20	
DUE RESEARCH			322.79	
ACCRUED ACCOUNTS				
Salaries and Commissions		$6,790.15		
Expenses		800.00	7,590.15	
DEFERRED INCOME				
Prepaid Membership Dues:				
Elected Members	$1,179.10			
Candidates	614.50	1,793.60		
Prepaid Initiation Fees		1,701.94		
Past President's Award		100.00	3,595.54	
REAL ESTATE MORTGAGE PAYABLE				
Cleveland Trust Co.			7,000.00	
RESERVE FOR PUBLICATIONS				
Transactions—Volume 53		12,000.00		
Transactions—Volume 54		12,000.00	24,000.00	
TOTAL LIABILITIES				$43,032.68
FUNDS				
Property Fund		83,439.17		
LESS: Mortgage Payable		7,000.00	76,439.17	
Reserve Fund			50,000.00	
F. Paul Anderson Fund			1,097.50	
Mortgage Reduction Fund			162.78	
General Fund			60,764.70	
NET WORTH OF SOCIETY				188,464.15
				$231,496.83

NOTE: This Balance Sheet is subject to possible assessments for social security taxes pending decision by the Treasury Department.

CONDENSED COMPARATIVE CONSOLIDATED BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—NEW YORK, N. Y., October 31, 1948

	October 31, 1948	October 31, 1947	Increases (Decreases)
ASSETS			
Cash on Hand and on Deposit	$88,878.18	$65,296.10	$23,582.08
Securities (At cost)	83,683.64	81,631.20	2,052.44
Accounts Receivable (Less: Reserve for Doubtful)	15,591.09	17,293.47	(1,702.38)
Inventory of Paper, Emblems, etc.	5,213.26	10,720.78	(5,507.52)
Deposits Receivable	425.00	425.00	
Accommodations	2,750.99	409.75	2,341.24
Mortgage Receivable	–0–	6,706.56	(6,706.56)
Land and Buildings (Less: Reserve for Depreciation)	81,012.89	82,459.31	(1,446.42)
Laboratory equipment, tools, furniture, fixtures and library (Less: Reserve for Depreciation)	23,002.52	20,721.38	2,281.14
Deferred Charges	2,595.65	3,682.14	(1,086.49)
TOTAL ASSETS	$303,153.22	$289,345.69	$13,807.53
LIABILITIES			
Accounts Payable	$ –0–	$11,935.14	$(11,935.14)
Federal Withholding Taxes	524.20	1,150.52	(626.32)
Accrued Accounts	7,998.26	7,045.44	952.82
Deferred Incomes:			
1947 Exposition	–0–	8,598.69	(8,598.69)
Research Projects	25,722.89	10,018.86	15,704.03
Prepaid Dues, Initiation Fees, Etc.	3,595.54	2,796.99	798.55
Real Estate Mortgage Payable	7,000.00	20,125.00	(13,125.00)
Reserve for Publication of Transactions	24,000.00	12,000.00	12,000.00
TOTAL LIABILITIES	68,840.89	73,670.64	(4,829.75)
FUNDS			
Property Funds	97,015.41	–0–	97,015.41
Reserve Funds	54,000.00	54,000.00	–0–
F. Paul Anderson Fund	1,097.50	1,072.11	25.39
Mortgage Reduction Fund	162.78	8,720.98	(8,558.20)
Research Endowment Fund	649.41	633.67	15.74
Research Fund	20,622.53	21,376.74	(754.21)
General Fund	60,764.70	129,871.55	(69,106.85)
Net Worth	234,312.33	215,675.05	18,637.28
TOTAL LIABILITIES AND FUNDS	$303,153.22	$289,345.69	$13,807.53

NOTE: The Property Funds were Created by Resolution of council on October 4, 1948.

COMPARATIVE CASH BUDGET SUMMARY

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—NEW YORK, N. Y. For the Fiscal Year ended October 31, 1948.

	Actual	Budget Provision	Increases	Decreases
RECEIPTS				
Dues	$104,986.13	$107,346.35		$2,360.22
Initiation Fees	7,621.94	8,100.00		478.06
Publications	134,146.65	132,300.00	$1,846.65	
Interest	785.70	750.00	35.70	
Other	12,367.17	6,200.00	6,167.17	
Research	104,783.86	103,532.55	1,251.31	
TOTAL RECEIPTS	$364,691.45	$358,228.90	$9,300.83	$2,838.28
DISBURSEMENTS				
Committees and Chapters	$ 21,094.41	$ 21,100.00	$	$ 5.59
Meetings	4,686.80	4,400.00	286.80	
Publications	87,776.92	103,100.00		15,323.08
Headquarters	91,832.29	87,150.00	4,682.29	
Mortgage Retirement	13,125.00	12,220.98	904.02	
Research	144,482.65	144,222.75	259.90	
TOTAL DISBURSEMENTS	$362,998.07	$372,193.73	$6,133.01	$15,328.67
DIFFERENCE		$ 13,964.83		
LESS: Balance in Mortgage Retirement Fund, October 31, 1947		$ 8,720.98		
Reserve for transaction for 1947		6,000.00		
		14,720.98		
SURPLUS	$ 1,693.38	$ 756.15		

BUDGET COMPARISON—CASH RECEIPTS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—NEW YORK, N. Y.

For the Fiscal Year ended October 31, 1948.

	Actual	Budget Provision	Increases	Decreases
DUES				
CURRENT MEMBERS				
100—Members	$ 45,655.12	$ 45,000.00	$ 655.12	
101—Associates	40,385.60	42,000.00		$ 1,614.40
102—Juniors	3,126.50	3,500.00		373.50
103—Students	136.00	500.00		364.00
	89,303.22	91,000.00	655.12	2,351.90
NEW MEMBERS				
104—Members	2,165.25	4,000.00		1,834.75
105—Associates	5,429.56	5,000.00	429.56	
106—Juniors	1,507.00	500.00	1,007.00	
107—Students	829.00	500.00	329.00	
	9,930.81	10,000.00	1,765.56	1,834.75
PRIOR YEAR'S DUES				
6A—Members	1,934.00	2,000.00		66.00
6B—Associates	1,931.50	2,500.00		568.50
6C—Juniors	93.00	150.00		57.00
6D—Students	-0-	-0-		-0-
	3,958.50	4,650.00	-0-	691.50
TOTAL DUES	103,192.53	105,650.00	2,420.68	4,878.15
108—INITIATION FEES	5,920.00	7,000.00		1,080.00
DEFERRED INCOME				
Prepaid Membership Dues	1,179.10	1,345.35		166.25
Prepaid Dues by Candidates	614.50	351.00	263.50	
108—Prepaid Initiation Fees	1,701.94	1,100.00	601.94	
	3,495.54	2,796.35	865.44	166.25
PUBLICATIONS				
115—Editorial Contract	20,166.64	20,500.00		333.36
300—Guide Advertising	54,512.95	50,000.00	4,512.95	
302—Guide Copy Sales	55,950.46	60,000.00		4,049.54
117—Transactions	1,089.59	1,000.00	89.59	
118—Books, Reprints and Codes	2,427.01	800.00	1,627.01	
	134,146.65	132,300.00	6,229.55	4,382.90
INVESTMENTS INCOME				
125—Interest—Savings Bank	67.69	200.00		132.31
126—Interest—Securities	475.05	300.00	175.05	
127—Interest—Mortgage	242.96	250.00		7.04
	785.70	750.00	175.05	139.35
OTHER INCOME				
109—Sales of Emblems	$ 1,051.62	$ 1,000.00	$ 51.62	
110—Sales of Certificates and Frames	590.32	500.00	90.32	
81—Mortgage Retirement	2,489.84	3,500.00		1,010.16
Mortgage Receivable	6,706.56		6,706.56	
82—Business Policy Committee	1,178.83	1,200.00		21.17
206—Past President's Award	100.00		100.00	
224—Refund of Pension Fund Deposit	250.00		250.00	
	12,367.17	6,200.00	7,198.50	1,031.33
RESEARCH LABORATORY INCOME				
34—Contributions—Earmarked	62,727.00	45,000.00	17,727.00	
33—Contributions—General	9,530.00	6,000.00	3,530.00	
Naval Research Contract		19,900.00		19,900.00
82—1948 Exposition Income	13,902.62	14,000.00		97.38
Deferred 1947 Exposition Income "A"	8,598.69	8,598.69		
Deferred 1947 Earmarked Contributions	10,018.86	10,018.86		
3A—Interest	6.69	15.00		8.31
	104,783.86	103,532.55	21,257.00	20,005.69
TOTAL RECEIPTS	$364,691.45	$358,228.90	$38,146.22	$31,683.67

NOTE "A"—The sum of $8,598.69 representing deferred 1947 Exposition Income is not to be considered as a cash receipt since it was collected in the prior fiscal period. It has been included only as a balancing figure herein and for the purpose of conforming with the budget.

BUDGET COMPARISON—CASH DISBURSEMENTS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—NEW YORK, N. Y., October 31, 1948

	Actual	Budget Provision	Increases	Decreases
COMMITTEES AND CHAPTERS				
150—President's Fund	$ 2,922.93	$ 2,500.00	$ 422.93	
151—Council Travel	5,151.06	6,000.00		848.04
160—Executive Committee	105.93	100.00	5.93	
161—Finance Committee	90.29	100.00		9.71
162—Membership Committee	828.80	1,000.00		171.20
164—Standards Committee	15.89	150.00		134.11
170—Admission and Advancement Committee	526.35	600.00		73.65
171—Constitution and By-Laws	692.68	1,500.00		807.32
172—Nominating Committee	1,021.37	2,000.00		978.63
173C—Chapter Relations Committee	393.38	300.00	93.38	
173A—Chapter Speakers	2,344.94	2,000.00	344.94	
173B—Chapter Delegates Travel	3,937.35	4,000.00		62.65
201—A.S.A. Membership	100.00	100.00		
109—Emblems	1,423.19	-0-	1,423.19	
204—Membership Certificates and Frames	1,330.45	500.00	830.45	
206—Medals and Awards	209.80	250.00		40.20
	21,094.41	21,100.00	3,120.82	3,126.41
MEETINGS				
163—Meetings	3,786.80	3,500.00	286.80	
327—Chapter Meeting Allowance	900.00	900.00		
	4,686.80	4,400.00	286.80	-0-
PUBLICATIONS				
200—Journal Subscriptions	13,282.68	13,500.00		217.32
202—Transactions—Volume 52	8,750.39	9,600.00		849.61
202—Transactions Volume 53		12,000.00		12,000.00
203—Membership Roll	4,115.24	2,000.00	2,115.24	
118—Books, Reprints, etc.	2,082.23		2,082.23	
305—Guide Publication and Distrib	59,546.38	66,000.00		6,453.62
	87,776.92	103,100.00	4,197.47	19,520.55
HEADQUARTERS				
210—Salaries	59,009.44	58,500.00	509.44	
211—Public Relations Expense	5,304.34	5,000.00	304.34	
212—Travel—Secretary and Staff	2,047.18	2,000.00	47.18	
213—Rent and Light	4,638.44	4,800.00		161.56
214—Telephone	1,467.23	1,200.00	267.23	
215—Telegraph	526.64	500.00	26.64	
216—Postage	3,947.58	3,000.00	947.58	
217—General Printing	1,836.33	1,000.00	836.33	
218—Office Supplies	1,444.45	2,000.00		555.55
219—Address and Address Changes	239.60	300.00		60.40
220—Professional Services—Legal and Auditing	2,594.84	3,300.00		705.16
220A—Business Policy Committee	1,178.83	1,200.00		21.17
221—Bank Charges	55.69	100.00		44.31
11—Furniture and Fixtures	915.11	750.00	165.11	
223—General Office Expenses	2,399.97	1,500.00	899.97	
224—Pension Fund	4,226.62	2,000.00	2,226.62	
	91,832.29	87,150.00	6,230.44	1,548.15
31—MORTGAGE RETIREMENT	13,125.00	12,220.98	904.02	
RESEARCH				
30—40% of Members and Associates Dues	39,308.59	40,200.00		991.41
30A—40% of Dues Prepaid in 1947 "B"	490.20	490.20		
82—1948 Exposition Income	13,902.62	14,000.00		97.38
27A—Deferred—1947 Exposition Income	8,598.69	8,598.69		
34—Contributions—Earmarked	62,727.00	45,000.00	17,727.00	
33—Contributions—General	9,530.00	6,000.00	3,530.00	
Naval Research Contract	-0-	19,900.00		19,900.00
34A—Deferred 1947 Earmarked Contributions	10,018.86	10,018.86		
3A—Interest	6.69	15.00		8.31
	144,482.65	144,222.75	21,257.00	20,997.10
TOTALS	$362,998.07	$372,193.73	$35,996.55	$45,192.21

NOTE "B"—The sum of $490.20 representing 40% Dues prepaid in 1947 is not to be considered as a cash disbursement of the current fiscal period. It is included herein only as a balancing figure and in order to conform with the budget.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—
RESEARCH FUND—NEW YORK, N. Y.
October 31, 1948

ASSETS

RESEARCH FUND				
CASH				
ON DEPOSIT				
Treasurer's Account—Bankers Trust Co.		$32,412.85		
Director's Account—Cleveland Trust Co.		7,393.77		
Travel Account—Cleveland Trust Co.		500.00		
Thrift Account—Bank for Savings		451.51	$40,758.13	
IN TRANSIT			3,029.42	
ON HAND FOR DEPOSIT			2,669.40	
ON HAND			250.00	$46,706.95
ACCOUNTS RECEIVABLE				
40% Dues Receivable		2,162.45		
Less: 40% Prepaid Dues	$ 393.24			
Building Depreciation	1,446.42	1,839.66	322.79	
U. S. Navy Department			2,334.19	
Advances to Retirement Plan for Account of Employees			976.77	3,633.95
PERMANENT				
Laboratory Equipment, Furniture and Fixtures		31,393.78		
Less: Reserve for Depreciation		11,117.54	20,276.24	
Tools			300.00	20,576.24
DEFERRED CHARGES				
Unexpired Insurance Premiums				415.92
TOTAL RESEARCH FUND				71,333.06
ENDOWMENT FUND				
CASH ON DEPOSIT				
Bank for Savings				649.41
				$71,982.47

LIABILITIES AND FUNDS

LIABILITIES			
ACCRUED ACCOUNTS			
Interest	$ 39.38		
Expenses	368.73	$ 408.11	
DEFERRED INCOME			
Heat Flow Through Glass	14,738.75		
Radiant Heating	7,780.00		
Physiological Research	2,441.00		
Sound Energy Studies	574.39		
Heat Losses Due to Infiltration	188.75	25,722.89	
EXCHANGES		3.29	$26,134.29
FUNDS			
Property Fund		20,576.24	
Reserve Fund		4,000.00	
Endowment Fund		649.41	
General Fund		20,622.53	45,848.18
			$71,982.47

NOTE "A"—This Balance Sheet may be subject to possible assessments for social security taxes pending final decision by the Treasury Department.

BUDGET COMPARISON—CASH RECEIPTS AND DISBURSEMENTS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—
RESEARCH FUND—NEW YORK, N. Y.
For the Fiscal Year Ended October 31, 1948

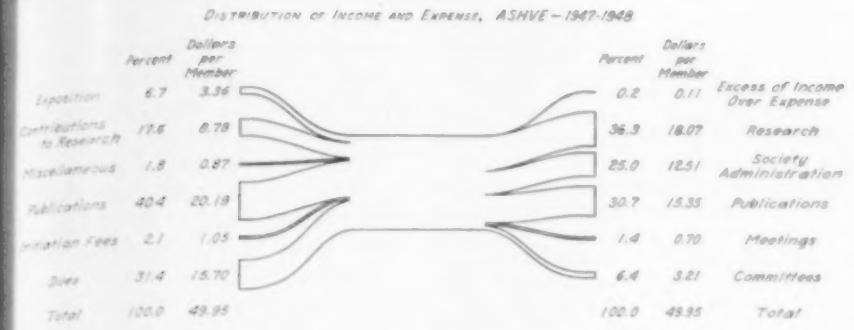
	Actual	Budget Provision	Increases	Decreases
CASH RECEIPTS				
40% Dues—Members and Associates	$ 39,208.59	$ 40,690.20	$	$ 1,481.61
Heating, and Ventilating Exposition—1948	13,902.62	14,000.00		97.38
Earmarked Contributions—1948 (A)	62,727.00	45,000.00	17,727.00	
General Contributions	9,530.00	6,000.00	3,530.00	
Interest	6.69	15.00		8.31
Deferred 1947 Exposition Receipts	8,598.69	8,598.69		
Office of Naval Research	-0-	19,900.00		19,900.00
TOTAL CASH RECEIPTS	133,973.59	134,203.89	21,257.00	21,487.30
Deferred 1947 Earmarked Contribution	10,018.86	10,018.86	-0-	-0-
TOTALS	$143,992.45	$144,222.75	$21,257.00	$21,487.30

EARMARKED CONTRIBUTIONS—1948 (A)				
Heat Loads and Heat Transfer including Solar Radiation Allied Studies	$26,550.00	$17,500.00	$ 9,050.00	
Physiological Research	5,882.00	6,000.00		118.00
Air Cleaning	7,550.00	6,000.00	1,550.00	
Air Distribution	1,765.00	2,000.00		235.00
Panel Heating Studies	17,780.00	10,000.00	7,780.00	
Oil Heat Institute Project	3,200.00	3,500.00		300.00
TOTALS (A)	$62,727.00	$45,000.00	$18,380.00	$653.00

NOTE—The budget provision of disbursements has been amended to include the supplementary budget of the office of Naval Research as follows:

100	—Travel of Committee	$ 1,500.00
150	—Technical Staff	3,865.00
150(A)	—Technical Staff Overhead	1,935.00
151	—Clerical Staff	400.00
151(A)	—Clerical Staff Overhead	700.00
160	—Travel Staff	1,000.00
160(A)	—Consulting Staff	500.00
208	—Building Alterations	5,000.00
300	—Equipment	5,000.00
	TOTAL	$19,900.00

	Actual	Budget Provision	Increases	Decreases
CASH DISBURSEMENTS				
100—Travel and Meeting Expenses, Chairman, Research Executive Committees and Technical Advisory Committees	$ 1,493.75	$ 3,150.00	$	$ 1,656.25
101—Chairman's Office Expense	153.57	200.00		46.43
	1,647.32	3,350.00		1,702.68
LABORATORY SALARIES				
150—Administrative and Technical Staff	51,078.99	54,115.00		3,036.01
150(A)—Technical Staff Overhead	-0-	1,935.00		1,935.00
151—Clerical Staff	8,882.49	9,150.00		267.51
151(A)—Clerical Staff Overhead	-0-	700.00		700.00
152—Pension Contributions	2,715.06	1,000.00	1,715.06	
153—Part Time Technical Staff	823.17	1,500.00		676.83
	63,499.71	68,400.00	1,715.06	6,615.35
LABORATORY EXPENSES				
160—Staff Travel	3,412.42	4,500.00		1,087.58
160(A)—Consulting Staff	-0-	500.00		500.00
161—Postage	589.02	650.00		60.98
162—Telephone and Telegraph	1,236.42	1,500.00		263.58
163—Office Supplies	1,020.85	1,000.00	20.85	
164—Mimeographing and Printing	1,306.42	1,250.00	56.42	
165—Library	246.21	250.00		3.79
166—Publications	1,712.39	2,500.00		787.61
167—Staff Insurance	126.51	200.00		73.49
168—Professional Services	250.00	250.00		
169—Unallocated	801.85	1,000.00		198.15
175—Contingencies	1,500.00	1,500.00		
250—Lab. Material and Supplies	3,819.96	4,000.00		180.04
300—Equipment (Permanent)	4,826.82	11,000.00		6,173.18
	20,848.87	30,100.00	77.27	9,328.40
LAB. OPERATION AND MAINTENANCE				
200—Mortgage Interest	585.29	675.00		89.71
202—Heat	1,027.68	1,200.00		172.32
203-5—Light, Power and Water	1,028.33	1,000.00	28.33	
206—Janitor Wages and Supplies	946.94	1,200.00		253.06
207—Building Maintenance	2,071.91	2,000.00	71.91	
208—Building Alterations	-0-	5,000.00		5,000.00
209—Building Insurance	217.65	200.00	17.65	
210—Legal Fees	1,759.67	1,800.00		40.33
	7,638.47	13,075.00	117.89	5,554.42
COOPERATIVE CONTRACT				
1000—Pledged to June 30, 1948	3,850.00	3,850.00		-0-
1100—July 1 to October 31, 1948	5,612.00	6,275.00		663.00
	9,462.00	10,125.00		663.00
	$103,096.37	$125,050.00	$1,910.22	$23,863.85



Report of Council

During the past year five meetings of the Council were held in New York, Cleveland, Bretton Woods, Memphis and Chicago. The Officers and Council Members spent many long hours in meetings and discussions of Society activities and this report will summarize some of the actions taken.

The organization meeting of the 1948 Council was held in New York on February 5, and President Tuve announced the personnel of the Council and Special Committees. Action was taken on the appointment of the Secretary of the Society, the Technical Secretary and the Assistant to the President.

The selection of depositories for Society funds was made and a certified public accountant was selected as required by the By-Laws.

During the year abstracts of Council Minutes were furnished to Chapter Delegates, Chapter Officers and others who desired to have them.

The question of the number of Society members in the various grades received special attention during the year and also a study was made of the membership qualifications required for various grades. A new Student Member application was also developed and approved.

Charters were granted for Student Branches at North Carolina State University and the Purdue Student Branch of Indiana Chapter and the organization of the Northeastern Oklahoma Chapter at Tulsa was completed in October.

The Council recommended to the Committee on Research an extension of the work in the physiological field and an increase in cooperative projects. Approval was given to the acceptance of Federal government sponsorship and support of research work in fundamental problems involving heating, ventilating and air conditioning, and a contract with the Navy was authorized.

An extensive survey of publication policy was made by a Special Committee and its recommendations were adopted.

Meeting programs for Bretton Woods and Chicago were approved and the invitation of the Minnesota Chapter to have the Semi-Annual Meeting 1949 at Minneapolis was accepted and the dates selected are June 20 to 22nd.

An invitation from the North Texas Chapter was approved for a meeting and Exposition at Dallas January 23 to 26, 1950 and the invitation of the Philadelphia Chapter for the Annual Meeting and Exposition in 1951 was also approved.

The Budget for 1949 was adopted and the recommendations of the Finance Committee regarding investment of Canadian funds and the liquidation of the mortgage on the Research Laboratory was approved.

New rates for advertising space in THE GUIDE are to be effective in 1950.

The suggestion of the Guide Publication Committee to eliminate the Roll of Membership from THE GUIDE 1949 was accepted and this will be published in a separate pamphlet which will include the names of the new officers and committees.

The resignation of Prof. F. W. Hutchinson as a member of the Committee on Research was accepted with regret and Prof. E. R. Queer was appointed to fill his term. Five candidates for election to the Committee on Research were nominated and the Council selected five members and an alternate to serve on the Nominating Committee.

The Council accepted a special contribution from Walter L. Fleisher, a past president of the Society for a Past President's award to one of the younger members of the Society who submitted the best manuscript on some subject dealing with environment.

A comprehensive study of Society administration and organization objectives was reported and will come up for future discussion.

The Public Relations work of the Society under the direction of the Public Relations Counsel was started on November 1 and a new editorial assistant for the Journal commenced his duties December 1st.

The Council took formal action required by the Constitution and By-Laws on the acceptance of resignations, submitted by Members and cancellation of memberships because of non-payment of dues.

The past 12 months have been notable for the many projects that have been started in the interest of expanding the Society's services and in extending its influence.

Respectfully submitted,
THE COUNCIL

Report of the Secretary

This year has been unique in the Society's history because it produced the greatest number of applications for membership ever recorded; total 1469.

The Admission and Advancement Committee was required to hold meetings almost every month in order to

NEW COUNCIL MEMBERS

S. W. Boyd
Atlanta, Ga.

C. R. Gardner
Dallas, Tex.

J. D. Kroeker
Portland, Ore.

C. O. Mackey
Ithaca, N. Y.

clear the requests for membership. The Membership Committee produced a new booklet on the Aims and Activities of the Society and, due to the combined efforts of the Membership Committee and the Chapter Relations Committee, there are several Chapters and Student Branches who are interested in permanent organization.

The report of the Finance Committee indicates that in dollar volume the Society reached another peak in its activities and with the adoption of the new Budget of nearly $450,000 for 1949 the growth of the Society in its 55-year history can be visualized.

The Society's publications represent a substantial percentage of the Society's work and with the growth of membership, the distribution of the JOURNAL, THE GUIDE and TRANSACTIONS has been greatly expanded. Over 18,000 copies of THE GUIDE 1948 were printed and only a few hundred are still on hand. The 1949 text section has been expanded by about 90 pages of entirely new material and a greater number of manufacturers are represented in the catalog data section.

Commencing with 1948, Amendments to the By-Laws required a mandatory schedule covering dues payments, and for the calendar year just past, collections were about 97 percent. At the Semi-Annual Meeting an increased dues rate for 1949 was adopted and payments have been promptly made by a large percentage of members.

The Secretary's office has handled all matters pertaining to meetings, production of publications, the collection of accounts, extensive correspondence with Officers, Committees and Chapters, operation of the Chapter Speakers Bureau, establishment of the Public Relations program and has carried on all of the routine actions required by the Constitution and By-Laws in connection with membership, finance, contracts and other special projects.

It is indicated in the Council report that a number of new surveys and projects have been started during the year, some of which were most essential because of the rapid growth of the Society and the necessity for trying to coordinate some of its anticipated activities. All of the work has been carried on with only nominal increase in the size of the headquarters staff and it is my pleasure to tell you that without their full cooperation, the work of the Society could not be handled.

The interest and assistance of the Officers, Council, Committees, and Chapter Officers during this year has been most helpful in carrying out the complex duties required of the Secretary's office and the staff is glad to acknowledge this cooperation.

Respectfully submitted,
A. V. HUTCHINSON,
SECRETARY

The Technical Session

The first paper, read by Nathaniel Glickman, **Physiological Adjustments of Normal Subjects and Cardiac Patients to Sudden Change in Environment,** by Nathaniel Glickman, Tohru Inouye, Robert W. Keeton, M.D., Irwin R. Callen M.D., Ford K. Hick, M.D., and Maurice K. Fahnestock (published in ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning,* February 1949) described tests on 16 cardiac patients and 10 normal subjects regarding their reactions to exposure to sudden changes in going from a comfortable room to a hot room and back to a comfortable room. Responses of cardiac patients, the paper stated, did not vary greatly from those of normal subjects.

The second paper, read by G. V. Parmelee, **Overall Coefficients for Flat Glass Determined Under Natural Weather Conditions,** by George V. Parmelee and Warren W. Aubele, (published in ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning,* December 1948) described tests made on sheets of glass exposed to wind velocities and also sheltered from the wind. Mr. Parmelee pointed out that computation of the radiation exchange and the convection loss due to turbulent wind conditions showed a need for improved instrumentation. A need was also expressed for field studies of heat loss through openings.

Welcome Luncheon

The official welcome at Monday's luncheon was offered by C. M. Burnam, Jr., president of the Illinois Chapter, who called on several guests at the speaker's table to rise and receive the applause of the audience. P. J. Marschall, treasurer of the Illinois Chapter, introduced the luncheon speaker, Dr. Henry T. Heald, who spoke on *The Professional Man and His Government.*

Dr. Heald described the tremendous increase in the student population since the war, with particular reference to higher education. Forecasting a college and university enrollment of three million students by 1960, Dr. Heald stated that local and state authorities would be unable to bear the financial burden and that the federal government would be required to furnish additional aid.

But the problem, said Dr. Heald, was how to satisfy the need for aid to higher education without having too much federal control. This problem invited the participation of thoughtful, professional people, such as the engineering profession provides. Engineers, he said, should be informed, taking time to participate in local, state and national problems, for the good of all.

(L. to R.) **W. A. Kuechenberg, general chairman, John Howatt, toastmaster, J. H. Milliken, vice chairman, and O. J. Prentice, publicity chairman**

Informal Party

In contrast to the serious tone of the morning's technical session, Monday night's informal party saw members relaxing and dancing to Johnny Jones and his orchestra. Outstanding in the entertainment were "Professor" Russell Oaks, inventor, and John O'Brien, harmonica player and comic. The professor displayed some remarkable inventions, including a butter cage for those wishing to reach for food across a table. Another amazing device was the hydraulic cigarette lighter which would have made Rube Goldberg very proud. In the event that the hydraulic device failed to work, the professor provided a switch-operated trap which produced a package of matches.

Johnny O'Brien is the comedian who believes that almost anything is possible, that's possible, and his use of this expression began to grow on the audience to such an extent that he was called back for repeated encores. But Mr. O'Brien did not mind, because he stoutly maintained that the spotlight to which he repeatedly returned was good for his chest.

Tuesday Technical Session

With registration proceeding busily, the Tuesday session was presided over by 1st Vice Pres. A. E. Stacey, Jr. After a report from the Tellers of Election, N. W. Snyder read his paper on **The Internal Performance of an Induced Draft, Packed Cooling Tower** (published in ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, February 1949). Professor Snyder reported on data obtained in 1942-43, when it was decided to check the performance within a cooling tower rather than the overall performance, as had been previously the case. Following discussion from the floor, Professor Snyder gave the closing remarks.

L. G. Seigel read a paper on **Heat Transfer Rates for Refrigerant Boiling in Horizontal Tube Evaporators**, by L. G. Seigel, W. L. Bryan and M. C. Huppert (published in the ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, January, 1949). The purpose of the investigation was to examine and correlate the factors which affected heat transfer significantly. The presence of oil in the evaporators, Professor Seigel stated, does not seem detrimental to heat transfer when the oil is present in quantities such as are obtained in commercial shipment. The amount of oil to be tolerated, however, should be determined by quantitative tests, Professor Seigel said. Closure of the paper was given by Mr. Bryan.

The subject of C. M. Ashley's paper, read by the author, was **Psychrometric Factors in the Air Conditioning Estimate,** (see page 107). Mr. Ashley described a method for selecting equipment and estimating air conditioning load. In this method, he used certain factors such as the sensible heat and bypass factors, and alluded to the appendices which showed how the equations were derived.

Luncheon Meeting

At Tuesday's Luncheon Meeting, Cyril Tasker spoke regarding his recent trip to England. Introduced by L. N. Hunter, chairman of the Committee on Research, Mr. Tasker stated that a great deal of home building is now going on in England, with several ministries participating in the setting up of

Illinois Chapter Hosts at Chicago Annual Meeting ***(L. to R.)*** **M. J. Bamond, G. W. Bornquist, G. V. Zintel, C. M. Burnam, Jr.**

Report of Tellers of Election

Ballot for Officers

	Total
President, A. E. Stacey, Jr., Syracuse, N. Y.	1453
1st Vice Pres., Lester T. Avery, Cleveland, Ohio	1454
2nd Vice Pres., L. E. Seeley, Durham, N. H.	1452
Treasurer, Ernest Szekely, Milwaukee, Wis.	1454

Members of Council

S. W. Boyd, Atlanta, Ga.	1366
C. R. Gardner, Dallas, Texas	1373
J. D. Kroeker, Portland, Ore.	1375
C. O. Mackey, Ithaca, N. Y.	1370
Total Ballots Received	1572
Total Legal Ballots	1470
Invalid Ballots	102

Committee on Research

3-year term

	Total
I. W. Cotton, Indianapolis, Ind.	1372
Nathaniel Glickman, Chicago, Ill.	1353
W. A. Grant, Syracuse, N. Y.	1367
L. N. Hunter, Johnstown, Pa.	1368
R. D. Madison, Buffalo, N. Y.	1360

2-year term

E. R. Queer, State College, Pa.	1407

Scattering votes:

President	5
2nd Vice President	1
1st Vice President	3
Treasurer	1
Council	5
Committee on Research	6

Board of Tellers
ROSWELL FARNHAM, *Chairman*
D. B. ANDERSON
B. S. FOSS, JR.

standards for the building and heating of these houses.

Pointing out the need for liaison between the United States and England, Mr. Tasker said that English engineers were very interested in the cooperative research now being conducted by the Society.

He advocated an exchange of engineering ideas and reports between the two countries and described the warm air heating studies now in progress on 20 English residences. In these studies, Mr. Tasker said, the subjective reaction of the tenants in the 20 homes would be noted in addition to the physical and physiological measurements.

Past President's Dinner

Tuesday was also the day of the traditional dinner given for the Society's past presidents. Dr. Baldwin M. Woods presided over the dinner, which was attended by the following: Homer Addams (the only living charter member), E. Holt Gurney, Harry M. Hart, Walter L. Fleisher, Frank B. Rowley, John Howatt, G. L. Larson, Merrill F. Blankin, W. H. Driscoll and A. C. Willard. S. H. Downs was unable to attend but met with the group later.

Wednesday Technical Sessions

Wednesday was a busy day, with two technical sessions.

During the morning session, presided over by Lester T. Avery, three papers on domestic heating were read. In the first paper, **Are Automatic Air Shutters Justified on a Gas-Fired Conversion Burner?** by W. M. Myler, Jr., and H. W. Nelson, published in the ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, October, 1948), Mr. Myler reported on a study made of an automatic air shutter on a gas-fired conversion burner. Two series of tests were run and results checked satisfactorily, with the conclusion that shutters are not justified on the type of burners used in the tests.

During the discussion on domestic heating, L. B. Schmitt read a paper on the **Performance of Residential Chimneys**, by L. B. Schmitt and R. B. Engdahl (published in the ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, November, 1948). The investigators found that type of construction or the materials used did not affect chimney performance appreciably. The authors also described a method for calculating the available draft of the small residential chimney.

Whereas the study read by Mr. Schmitt dealt with two chimneys located inside a building, the paper **Performance of Fourteen Masonry Chimneys Under Steady State Conditions**, by P. R. Achenbach and S. D. Cole (published in the ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, October, 1948) dealt with fourteen chimneys which were partly exposed above the roof for one-third of the height. Results of both papers, however, were in close agreement according to Mr. Achenbach, and also according to Mr. Engdahl. Further, the results obtained for brick chimneys did not differ greatly from those obtained

Exhibits in the Arena of the International Amphitheater

for chimneys made of shale-tile and cinder-concrete, Mr. Achenbach said.

At the afternoon session, R. J. Lorenzi read the first paper, **Performance of an Electrical System of Panel Heating with Four Stages of Insulation**, by R. J. Lorenzi and J. F. Schreiber (published in the ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, January, 1949) describing a four-room home which was completely heated by panels of electrically conductive rubber. Reduction of heat loss was noted after the addition of insulation to ceiling and floor in successive stages.

Heating a Basementless House with Radiant Baseboard, by R. H. Weigel and W. S. Harris (published in ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, November, 1948) was the paper read by Mr. Weigel.

The hot water heating system used responded quickly to sudden changes in load, thus maintaining a constant room air temperature, the paper stated, and comfortable floor slab temperatures were readily obtained in the basementless structure.

Continuing the session on home heating, Ben Ciscel read a paper entitled **Heating Panel Time Response Study**, by Mr. Ciscel and A. B. Algren (see p. 103.). The purpose of the study was to obtain basic information on automatic control of floor panel heating systems. Mr. Ciscel pointed out that the tests indicated that the mean radiant temperature was generally within a degree of the room air temperature.

During this session, C. M. Humphreys, senior engineer of the ASHVE Laboratory, read the progress report of Group A of the Technical Advisory Committee on Panel Heating and Cooling. Beginning in the latter part of 1947, the program was directed by the TAC with a sub-group of A, B, and D committees. By the end of January 1948, Mr. Humphreys said, the literature in the field of panel heating had been surveyed and two concrete slabs were poured. He said that although no conclusive results were yet available, progress is being made.

Other Highlights

In between technical sessions, council, committee and chapter delegates meetings, members of ASHVE still found time to participate in industrial plant tours.

A group of 69 availed themselves of an opportunity to visit the merchandise testing and development laboratories at Sears Roebuck & Co. The visitors were shown the workings of the various testing activities. Some of the more outstanding items were: the anechoic chamber used for the finer measurements of sound in connection with radio equipment; the home economics laboratory with numerous and ingenious machines for performing such tests as batting baseballs and testing mattresses.

400 Ladies Entertained

By Mrs. Chas. E. Price, Committee Secretary

This is a report of the ladies activities, under the able direction of Mr. Joseph S. Kearney and Mrs. W. A. Kuechenberg. It was the aim of this committee to meet and really welcome every visiting lady. Did you know there were almost 400?

The Get-Together Lounge was always in charge of at least two hostesses, who were ready to greet each visitor, and answer any questions about places to see and things to do in Chicago. This Lounge was really dedicated on Monday afternoon, when an informal reception was held there. It was very well attended, and all those present seemed to enjoy the friendly air and the punch bowl.

Tuesday noon, buses were provided to take wives and daughters of members to the Edgewater Beach Hotel, where they were treated to a delicious luncheon, and entertained by a fashion show. This was provided by students of the Ray-Vogue School of Design. Mrs. Kay Morrow, an instructor at the school, who is also well-known in radio and television work, talked on fashion trends and personal fashion problems.

Wednesday morning, the hardy got up at dawn to attend the widely known broadcast of the "Breakfast Club." Mrs. S. E. Fenstermaker, Carmel, Ind., and Mrs. Reg F. Taylor, Houston, Tex., were called upon by Don McNeill to speak to the radio audience. After the show, the ladies were entertained at a breakfast party in the Merchants and Manufacturers Club in the Merchandise Mart. Professional guides were on hand to take groups on a tour of the Merchandise Mart, and stops were made in typical salesrooms on each floor visited.

All those who attended the informal party on Monday night and the banquet on Wednesday night felt that those affairs were most successful.

The Ladies Committee worked hard to give their visitors a happy time in Chicago, and if it was work, it was of the most pleasant and rewarding kind. They made lots of new and delightful friends and greeted old familiar ones. They hope the visitors enjoyed themselves too, and will want to come back to Chicago soon again.

As these words are being written, the sun shines in Chicago! Will anyone believe that statement?

Annual Banquet

High point of the 55th Annual Meeting was the Banquet, attended by 1100 members, ladies and guests. Toastmaster John Howatt introduced Homer Addams, the only charter member of the Society, to the audience and also read messages of regret from Past Presidents C. V. Haynes, S. R. Lewis and Dr. F. E. Giesecke at being unable to attend.

Dr. B. M. Woods was given a memory book on behalf of the Society with the presentation being made by toastmaster John Howatt. A past president's emblem given by the Society was bestowed on Prof. G. L. Tuve by Prof. G. L. Larson.

W. A. Kuechenberg introduced Dr. Clark G. Kuebler, the banquet speaker, whose subject was *A Formula for Freedom*. Freedom is not easily won, Dr. Kuebler said, because "freedom has never been free." He said that World War II was one phase of a struggle that is still going on—"a battle between two opposing philosophies of life," that of enslaving the people, or that of

(L. to R.) G. L. Tuve, Mrs. A. E. Stacey, Prof. G. L. Larson

(L. to R.) Mrs. G. L. Larson, A. E. Stacey, Jr., Mrs. G. L. Tuve
At the Head Table

Annual Banquet with the Texas delegation in foreground

emancipating them. America should be governed by discussion and not by blind submission, Dr. Kuebler emphasized, and illustrated his point by citing his experiences in Nazi Germany, particularly his witnessing proceedings in a Nazi court.

In the struggle between opposing ideologies, America requires each of us to discharge responsibilities as well as to enjoy privileges of democracy, the speaker said. Engineers should use their trained thinking to uphold democracy and maintain equality of opportunity for all.

An invitation to attend the Semi-Annual meeting, June 20 to 22 in Minneapolis, was extended by John Haines of the Minnesota Chapter, and an enthusiastic Texas contingent led by C. Rollins Gardner, invited the members to come to Dallas for the 56th Annual Meeting.

The guests at the speaker's table were: Mr. and Mrs. W. A. Kuechenberg, Mr. and Mrs. C. M. Burnam, Jr., Mr. and Mrs. Ernest Szekely, Mr. and Mrs. C. F. Roth, Mr. and Mrs. Lester T. Avery, Dean and Mrs. L. E. Seeley, Mr. and Mrs. A. E. Stacey, Jr., Prof. and Mrs. G. L. Larson, Prof. and Mrs. G. L. Tuve, Dr. Clark G. Kuebler, Mr. and Mrs. John Howatt, and Dr. B. M. Woods.

Thursday Technical Session

H. B. Nottage, who read his paper on **Turbulence—A Fundamental Frontier in Air Distribution,** (to be published in the ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*) emphasized that basic data were needed regarding this subject, which has been incompletely understood. The study of the mechanisms of turbulence remains a frontier, Mr. Nottage said—a frontier which is being explored in a long-range project. However, certain engineering applications were indicated for dealing with friction, turbulent diffusion, and free jets.

Balancing Air Delivery of a System of Manifold Air Diffusers, by G. S. Dauphinee and Peter Argentieri (to be published in the ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*), was outlined by Mr. Argentieri. In this study, the data which enabled the authors to bal-

ance the system by the use of properly designed and calibrated dampers are tabulated.

The paper, **Air Distribution and Draft**, by John Rydberg and Per Norbäck, (published in the ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, January, 1949) was read by C. M. Humphreys. The paper discusses the theory of air flow and temperature distribution in jets. Formulas are given for determining the increase in air temperature necessary to compensate for the velocity of air impinging on occupants of air conditioned rooms.

Chairman Tuve then turned the meeting over to W. H. Driscoll who presided over the installation of newly elected officers.

Retiring President Tuve then turned the gavel over to the incoming president, A. E. Stacey, Jr., and the 55th Annual Meeting passed into the pages of Society history.

Resolutions

WHEREAS, Prof. G. L. Tuve has served the Society long, efficiently and faithfully on many committees, the Council, and as Vice President and President,

WHEREAS, his very able application of his talent to Society affairs, particularly research, has contributed so greatly to the advancement of the art of heating and ventilation,

BE IT RESOLVED, That the ASHVE extends its most sincere thanks to him for his generous and unselfish efforts in behalf of this Society.

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WHEREAS, this, the 55th Annual Meeting of the Society, has been one of the most outstanding Annual Meetings that the Society has ever held, not only in the all-time record attendance, but also in the quality and interest in the papers presented, in the work accomplished at the many committee meetings, in the enjoyable entertainment provided for the ladies, the members and the guests, and in the number and attraction of the exhibits at the 9th International Heating and Ventilating Exposition,

BE IT RESOLVED, That the appreciation and sincere thanks of the members of the Society be extended:

To the Illinois Chapter for the splendid manner in which they planned, organized and conducted the Annual Meeting.

To the Officers, the Council and the Staff Executives of the Society for their able direction of the Annual Meeting.

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To the authors of the various technical papers presented at the Annual Meeting, for the knowledge, thought and time which they contributed in preparing and presenting their papers.

To the speakers who addressed the meeting, for their interesting and informative talks.

To those who so pleasantly entertained the members at different times during the meeting.

To the daily newspapers and the technical press for the fine reporting of the meeting.

To the management and staff of the Chicago hotels, who provided the hotel facilities for the Annual Meeting, for the splendid service and excellent cooperation.

To the manufacturers and the exhibitors at the Heating and Ventilating Exposition for the excellent displays and to Charles F. Roth and his staff for the efficient handling of the Exposition.

To the railroads and airways for assistance in transportation.

Respectfully submitted,
LORIN G. MILLER, *Chairman*
M. C. BEMAN
G. LORNE WIGGS

The Twin Cities Members

of

MINNESOTA CHAPTER

cordially invite all members of the Society to attend

THE SEMI-ANNUAL MEETING 1949

June 20 - 22

Hotel Nicollet, Minneapolis, Minn.

O. L. Lilja, *President*
Minnesota Chapter

John E. Haines, *Chairman*
Committee on Arrangements

During the tour of the Ilg Electric Ventilating Co. plant, 84 members and guests witnessed an automatic dipping and insulating process for motor insulation, as well as a process whereby metal parts were washed and dried automatically. Other automatic machines viewed were a press which stamped out 150 laminations per minute and a machine which stamped out unit heater fins from coiled copper stock. At the firm's research laboratory, tests on a centrifugal fan were demonstrated.

More than 100 ASHVE members and guests turned out for the inspection trip of McDonnell & Miller, Inc., held on Thursday, January 27, at 1:30 p.m. and at other hours during the week. In the plant and laboratory, the visitors were shown the processes involved in the development, testing and manufacture of the company's products for controlling water level and protecting boilers against low water hazards.

Entering the plant of McDonnell & Miller, Inc., for inspection trip

Seen at the Exposition

Three hundred exhibitors . . . air conditioning units . . . humidifiers . . . gas, coal, oil unit heaters . . . blowers . . . automatic controls . . . registers . . . grilles . . . fans . . . pumps . . . coils . . . traps . . . valves . . . thermostats . . . control panels . . . evaporators. . . . . boilers . . . roof cooling system . . . relays . . . cut-offs . . . furnaces . . . conversion burners kitchen ventilators . . . air filters . . . precipitators . . . condensers . . . water heaters . . . piping . . . dehumidifiers . . . insulating materials . . . baseboard panels . . . copper tubes . . . fins . . . coolers . . .

Summary of Local Chapter Meetings

Kansas City—December 6, 1948. At the December meeting in the Pine Room, Union Station, Pres. H. E. Gould called on C. W. Schumacher to give a report on the projected Student Branch at the University of Kansas, Lawrence, Kans. He stated that temporary officers had been elected. Among them were President-elect Vance Elder, who spoke on the interest of the students.

Following a report by D. M. Allen on the Society's Annual Meeting next month, John James, McDonnell & Miller, Inc., Chicago, was introduced by L. E. Zumbehl. Mr. James described the development of steam boilers and also the development of the *ASME* code for rating relief valves for hot water systems. He emphasized safety precautions and the use of safety devices for various types of hot water heating systems. Consumer education is needed, he said, to acquaint users with the refinements that should be included in the heating systems. A question and answer period followed his talk. Attendance 82.

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Manitoba—December 9, 1948. A regular meeting was not held during December. In its place a mixed party was held in the form of a supper-dance at the Highwayman on Pembina Highway, and this affair was judged a success. Attendance 62.

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Massachusetts—December 15, 1948. The December meeting was the annual social meeting held jointly with the Boston section of *ASRE* in the Lotus Room of the University Club, Boston. In addition to 69 ASHVE members, 53 refrigerating engineers were present for a total of 122.

This joint meeting featured a social hour, dinner, and two guest speakers, Denny Myers and W. M. Scenna. Mr. Myers, head football coach of Boston College, spoke on the Holy Cross-Boston College football game. The subject of Mr. Scenna's talk was *Deep Sea Fishing*. Both talks were accompanied by movies and were thoroughly enjoyed by the audience. Attendance 122.

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Michigan—December 13, 1948. The December meeting at the Horace A. Rackham Memorial Building was a dinner held in conjunction with the *ASRE*. Eric Hyde, president of the Michigan Chapter, announced the offering of the refresher course by E.S.D. for engineers wishing to take the state examination for registered professional engineers.

Speaker of the evening was H. C. Hoffmann, of the Carrier Corp., Syracuse. Introduced by L. A. Burch, vice president of the Chapter and chairman of the joint meeting, Mr. Hoffmann gave a paper on the principles of the air conditioning equipment and distribution system as applied to a large hotel building in Cincinnati. He stated that the central system handles approximately one-quarter of the total air through the conditioning equipment and distribution system. Mr. Hoffmann compared various systems in terms of cost of controls and feeling of discomfort. Following the presentation of the paper, many questions were asked by the members. Attendance 77.

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Minnesota—December 6, 1948. The December meeting was held at the Curtis Hotel following a dinner at 6:30 p.m.

G. M. Kendrick of the program committee was introduced by O. L. Lilja as the person who had arranged for the speaker of the evening. This speaker was T. A. Walters, director of research for the American Blower Corp., Detroit. Mr. Walters gave an informative talk on the subject of *Fan Selection From An Acoustic Viewpoint*. He displayed special representative performance charts and discussed their proper use in selecting a fan. An interesting question and answer period was held at the conclusion of Mr. Walters' discussion.

Business conducted included reading the treasurer's report by Pres. O. L. Lilja, who presided over the meeting. L. J. Krause, chairman of the entertainment committee, was in charge of the program. He also announced the entertainment for the January meeting. Attendance 111.

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Montreal—January 10, 1949. The annual curling bonspiel was held at the St. George Curling Club. Curling started at 12:30 p.m. and dinner was held at 6:45 p.m. At 7:50, Pres. Leo Garneau called the meeting to order and then turned it over to B. J. Horsburgh, chairman of the golf and curling committee. Awards were given to the following teams: first award—C. Hennesey, S. R. Plamondon, W. J. Robinson and James Baker; runner up—Ivan McDonald, H. E. Blatherwick, John Reid, and J. F. Woodcock.

Thanks were given by the Chapter to the golf and curling committee which consisted of B. J. Horsburgh, chairman, Jack Wright, W. G. Hole and J. J. Cosgrove. Attendance 70.

Montreal—December 14, 1948. The Chapter's December meeting, held

at the Berkeley Hotel, was the annual Christmas party and Past Presidents' Night, with Pres. Leo Garneau presiding. Past Presidents' pins were presented to the following: G. Lorne Wiggs, F. J. Friedman, C. W. Johnson, F. A. Hamlet, A. B. Madden and T. H. Worthington.

J. J. Cosgrove was assisted by S. W. Salter in directing the entertainment. Presents were distributed by Mr. Cosgrove, disguised as Santa Claus. Attendance 64.

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New York—January 17, 1949. Ninety members and guests attended the meeting of New York Chapter on January 17th, at which Pres. A. A. Bearman presided.

C. S. Koehler, chairman of the membership committee, announced the names of 20 new Chapter members.

The following chapter nominating committee was appointed, with instructions to report at the February meeting: H. H. Bond, *Chairman* and C. S. Koehler, H. J. Ryan, W. A. Sherbrooke, and M. C. Giannini.

The subject *Solar Energy and the Air Conditioning Engineer* was presented by G. V. Parmelee, research fellow at the ASHVE Research Laboratory in Cleveland. Mr. Parmelee discussed the transmission of solar energy from the sun to the earth, and explained the special apparatus used at the Laboratory for measurement of the energy received by direct radiation from the sun, and by diffuse radiation from the sky.

Mr. Parmelee discussed the methods given in THE GUIDE for computing the heat transfer from the sun and from the outside air through glass walls and roofs. After illustrating the difference between periodic and steady state heat flow, and illustrating the difficulty of relating the instantaneous heat gain to the actual cooling load, the speaker explained the method of using simplified equivalent temperature differential tables prepared by J. P. Stewart, for determining the cooling load due to heat transfer through walls and roofs. In his talk, Mr. Parmelee referred frequently to the Cooling Load chapter of THE GUIDE, and showed the relation between the data collected at the ASHVE Research Laboratory and the tables published in THE GUIDE.

Mr. Parmelee's talk was greatly appreciated by the members because they felt that as a result of it they would now be able to make better use of the design information published in Chapter 15 of THE GUIDE. Attendance 90.

New York—December 14, 1948. The December meeting was in the form of a Christmas party, held in the Building Trades Club, New York City. A fine dinner, excellent entertainment, and dancing provided 300 members and guests with one of the most successful parties ever held by the Chapter.

Bernard Leventhal, chairman of the Christmas party committee, was in charge of the program and arrangements.

New York—November 15, 1948. Following a social hour and a dinner in the Building Trades Club, Pres. A. A. Bearman called upon membership chairman C. S. Koehler, who gave a membership report. President Bearman then read a notice of meetings of various engineering societies listed by the Engineering Societies Council of New York.

At the invitation of President Bearman, C. A. McKeeman, assistant to the president of ASHVE, spoke briefly on the progress of ASHVE research and mentioned the support being given the research program by industry and by ASHVE members.

Program chairman A. A. Giannini introduced Henry Sleik, vice president of W. B. Connor Engineering Corp., New York, who spoke on the subject: *Odors, Their Nature, Cause and Control.*

Mr. Sleik traced the attempts made in defining odors and outlined the methods used for odor control, such as ventilation and adsorption using activated carbon. He also showed slides of available equipment for adsorption of odors and described the use of the equipment. He referred to the potentialities of odor adsorption in contributing to the economy of the air conditioning systems.

At the end of the presentation of the subject, Mr. Sleik answered questions on the use of activated carbon equipment and on the requirements for keeping it in operating condition. Attendance 120.

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Northeastern Oklahoma—December 8, 1948. A period of fellowship and a dinner preceded the December meeting in the Red Lacquer Room of the Alvin Hotel, Tulsa. Several business matters were disposed of before the guest speaker, John James, was introduced by F. M. Thomas, chairman of the program committee. Mr. James spoke on *Recent Research Developments on Hot Water Heating* and answered questions during the discussion period which followed.

Chapter business conducted at this meeting included discussion and adoption of the by-laws recommended by the constitution and by-law committee and election of W. R. Lee as Chapter delegate and F. M. Thomas as alternate, to attend the Chicago Meeting. Boyd Purifoy requested that a committee be chosen to investigate the new proposed city code for refrigeration. President Lee read the proposed code to the Chapter and a lively discussion followed. A committee consisting of J. C. Penafeather, *Chairman*, T. E. Spangenberg and J. N. Watt, was appointed.

Presiding over the meeting was W. R. Lee, president. Attendance 26.

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Oklahoma — January 10, 1949. A dinner preceded the meeting at Beverly's Drive-In. Speaker of the evening was J. W. May, director of research, American Air Filter Co. Inc., Louisville, Ky. Mr. May gave an interesting talk on air filters, with emphasis on the role of electronics in air filtering equipment. He also showed slides of the different types of electronic precipitators and discussed the latest trends in air cleaning. During his talk Mr.

May was assisted by Dallas Deem. Pres. J. H. Spaan presided. Attendance 40.

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Ontario—January 3, 1949. Following dinner at the Royal York Hotel, Pres. E. G. Spall called the meeting to order. N. A. Smith reported on the progress on the Canadian Room committee, pointing out that the Canadian Room will be the reception room in the Stevens hotel for Canadians attending the 55th Annual Meeting. This room will be operated by a joint committee of representatives from Montreal, Toronto and Winnipeg Chapters.

Introduced by Robert Dunn, the speaker of the evening was H. A. Mosher, of the engineering department of Eastman Kodak Co., Rochester, N. Y. Mr. Mosher spoke on *When Is Air Conditioning of the Modern Factory Advisable?* Using slides to illustrate his talk, he dealt chiefly with the economic aspects of air conditioning in industrial plants, and discussed the types of plants where air conditioning is advisable and necessary. A discussion period followed Mr. Mosher's address. Attendance 95.

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Oregon—December 9, 1948. Members of the Oregon Chapter enjoyed the humor of Rev. John B. Delaunay, who spoke on his experiences in India, *In the Land of Heating and No Ventilating.*

Rev. Delaunay, head of the psychology department and also dean of men, Portland University, related that he had been sent to India for the good of the Church and later was returned to the United States for the good of India. For the balance of his nontechnical talk he detailed many of the differences in the likes and dislikes of the Hindus and Mohammedans; the result of befriending the natives; the difficulties encountered in ordinary living, and the ingenious expedient he developed to provide the visiting Bishop with a good dinner.

After a brief recess, the meeting was resumed for business. Pres. E. R. Lokey gave a report of actions taken at the previous board of governors meeting, including K. N. Flocke's report regarding the 6th Air Conditioning Conference. Mr. Flocke then gave a report to the chapter on this conference, which will be held March 24 to 26 at Oregon State College, Corvallis. These dates were approved by Prof. E. C. Willey.

President Lokey read a report of the membership committee, following which W. H. Stover gave a report on the legislative committee. He said that this committee has been conferring with the Oregon State Board of Health, and has recommended some changes in the regulations for industrial health. The forced warm air heating code was reported on by Walter Hanthorn.

Other business included reports by Treas. J. H. Bonebrake and, Chapter Delegate J. P. McDermott. A social committee report by J. A. Freeman set the Christmas party date for December 23, 1948. W. J. Widmer, chairman of the committee for mementoes for past presidents, stated that his committee recommended presentation of properly identified bookends to past presidents. This recommendation was unanimously approved by the chapter.

Pres. Lokey announced that the Council had elected W. B. Morrison a member of the ASHVE Nominating Committee. Mr. Widmer gave a report on the Society's TAC on Panel Heating and President Lokey then asked for volunteers to cover the panel heating committee meetings in Chicago. Mr. Morrison offered his services. Attendance: 72.

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Pacific Northwest — January 11, 1949. A debate on temperature controls marked the January meeting in the Engineers Club, Seattle, which was presided over by Pres. L. L. Bysom. The debate was conducted by three speakers, with E. L. Weber as referee. The speakers were: S. D. Peterson, of the Johnson Service Co., who presented the side of pneumatic controls; M. N. Musgrave, of Barber-Colman Co., who took the opposite side of electric controls; and P. B. Tack of the Minneapolis - Honeywell Regulator Co., who presented the main points of both electric and pneumatic controls.

After the three speakers had completed their presentations, an open discussion and question-and-answer period was conducted.

J. D. Sparks, chairman of the entertainment committee, was in charge of the program. A business discussion was held on the subject of the chapter's delegate to Chicago, and the subject of the next meeting was announced. Attendance 64.

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Pittsburgh—January 10, 1949. Research in the field of heating was discussed by L. N. Hunter. Speaking before the Chapter in Hotel Webster Hall in Pittsburgh, Mr. Hunter, vice president and chairman of research for the National Radiator Co., Johnstown, Pa., pointed out the difference between pure and applied research. He outlined the function of research committees and described some of the changes taking place in the study of heat. After his talk, Mr. Hunter answered questions from the audience.

During the business part of the meeting, B. B. Reilly gave the treasurer's report and E. C. Hach reported on memberships. H. J. Kirkendall of the program committee announced P. S. Park, Jr. of A. M. Byers Co., Pittsburgh, as the speaker for the February meeting. The meeting, said Mr. Kirkendall, would be held on the first Monday in February. T. F. Rockwell discussed aspects of the Society.

A dinner preceded the regular meeting, at which Pres. H. E. Park presided. President Park also introduced the speaker of the evening. Attendance 53.

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Rocky Mountain—December 8, 1948. Paul H. Wright was the dinner guest at the dinner meeting held at the Olin Hotel. Several business matters were brought up, including memberships, past presidents' pins,

and the code committee. Nelson H. Brickman reported that a rough draft of the proposed code would be completed. After revisions, he said, recommendations will be made to an advisory committee before the draft is brought to the City Council's attention. Chapter members voted to accept the recommendations of the code committee.

Pres. Fred Janssen, presiding over the meeting, announced the appointment of E. J. McEahern as delegate to the Chicago Meeting. President Janssen will act as alternate. The chapter voted expenses for the chapter delegate, and the meeting was then turned over to L. M. Hook of the program committee.

Mr. Hook introduced the guest speaker, R. L. Gordon, gas utilization engineer with the Public Service Co. of Colorado, who spoke on *The Effect of Altitude upon Combustion.*

Mr. Gordon stated that the composition of natural gas in Denver and vicinity is about 75 percent methane, 5 percent ethane, 15 percent nitrogen and 5 percent other gases, but no carbon monoxide. The gas has a heating value of 980 Btu per cubic foot referred to sea level conditions or about 830 Btu per cubic foot under Denver conditions. In combustion, 11.8 percent is the ultimate attainable carbon dioxide content of the flue gases with no excess air. Mr. Gordon cautioned that flue gas measurements of both oxygen and carbon dioxide should be made in order to avoid gross errors, possible when only carbon dioxide content is measured. The production of carbon monoxide will be accompanied by aldehydes which impart a sweetish odor to the flue gas. The kindling temperature of the gas-air mixture varies from 1100 to 1400 deg F and the maximum flame temperature is about 3500 deg F. Stack losses increase rapidly as the flame temperature approaches 3500 deg F unless an economizer is used. Natural gas has a speed of flame propagation of 1.2 fps as compared to 8 fps for hydrogen. The low rate of flame propagation decreases the possibility of *flashback* and increases the possibility of *blowoff* of the flame.

Air at the altitude of Denver contains only 85 percent as much oxygen per cubic foot as air at sea level. Mr. Gordon demonstrated this effect by means of apparatus including a pressure-vacuum pump, a pressure gage, and a spherical glass bell jar containing an inflated rubber balloon.

The heating capacities of gas appliances in Denver will, in general, be reduced about 20 percent below sea level conditions. Often minor changes will restore sea level rating. Design of appliances must provide for passage of larger gas-air mixtures in Denver. The efficiencies of gas appliances operating under high-altitude conditions will change very little, Mr. Gordon said.

Appliances to be used in Denver may be tested at sea level conditions by applying 126 percent of the Denver rating. The results of such a test will be comparable to results obtained when the appliance is operated at Denver conditions with a Denver rating applied.

An enthusiastic question and answer period followed the talk. Attendance 31.

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St. Louis—January 4, 1949. A number of business matters were taken up at the January meeting held in the York Hotel, following which Mr. Williamson of the Union Electric Power Co. spoke on fly ash.

Reports to the membership included: treasurer's report; report on attendance, by E. T. Clucas; membership report by H. R. Halt, chairman of the membership committee: and W. A. Russell reported on the February meeting.

F. E. Ince invited the chapter to attend the *ASRE* meeting at Bill Medarts on January 10. The speaker at the meeting, said Mr. Ince, would be W. S. Bodinus, district manager, Carrier Corp., Chicago, who would discuss *High Pressure Air Distribution.* Following a motion by Mr. Clucas, the chapter unanimously agreed to hold a dinner dance in the Spring.

Pres. B. L. Evans, presiding over the meeting, stated that the board of governors had been working on the rules and by-laws of the Chapter. J. H. Carter further pointed out the need for changes in them. The chapter then authorized the secretary to reproduce and mail a list of suggested rules and by-laws to the members.

Mr. Williamson's talk was illustrated by slides, and covered in detail the work of the electric company at the Cahokia power plant in reducing fly ash, which comprises 10 percent of the air pollution over the St. Louis area. The program for reduction of fly ash, described by Mr. Williamson, involved $4,000,000 directly for dust collectors and $12,000,000 indirectly for improvement of power plants. Non-residustrial sources of air pollution were also discussed by Mr. Williamson. A spirited question and answer period followed. Attendance 61.

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Shreveport—December 9, 1948. In the Colonial Room of the Caddo Hotel, chapter members elected the following candidates: A. H. Otto, *president;* B. E. Segall, Jr., *vice president;* S. W. Beaty, *secretary;* R. M. Hood, *treasurer.* P. O. Rottman, W. S. Evans, R. M. Hood, A. H. Otto, B. E. Segall, Jr., S. W. Beaty, and W. F. FitzGerald were elected to the *Board of Governors.*

Reports were made by the legislative, finance, membership, and publicity committees. President FitzGerald, who presided over the meeting, appointed the following to the auditing committee: B. E. Segall, Sr., R. M. Hood, and A. H. Otto.

The guest speaker was L. W. Cunningham, of Oakite Products Corp. Introduced by Mr. Otto, Mr. Cunningham spoke on *Cooling Tower Water Treatment.* After Mr. Cunningham's talk, Mr. Otto gave a brief discussion of why people attend ASHVE meetings. It was announced that J. W. May, Louisville, Ky., would be guest speaker for the January meeting. Mr. Segall was appointed chairman of the program committee. Attendance 39.

Shreveport—November 10, 1948. A talk on *Professional Air Control in 1960* was given by ASHVE Pres. G. L. Tuve at the dinner meeting held in the Washington-Youree Hotel. In addition to predictions as to the trend of air conditioning in the future, Professor Tuve made some comments on engineering as a profession. During the business part of the meeting, the following members were elected to the nominating committee: B. E. Segall, Sr., P. O. Rottman, S. W. Beaty, J. W. Brown, and A. H. Otto. Attendance 44.

Shreveport—October 23, 1948. Pres. W. E. FitzGerald called the meeting to order in the Colonial Room of the Caddo Hotel at 6:00 p.m. All business was dispensed with and the meeting was turned over to A. H. Otto, who introduced Harold Abbot, director of the Dale Carnegie School on Human Relations and Effective Speaking. Mr. Abbot, in spite of a bad throat, gave a most interesting talk.

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South Texas—December 17, 1948. Discussion of the Chicago meeting was one of the highlights of the December gathering in the Rice Hotel, Houston. C. R. Gardner of the North Texas Chapter gave a talk about the Annual Meeting, and a collection was made to help send members of the student branches at Texas A. & M. and the University of Texas to Chicago.

A discussion was led by B. P. Fisher relative to proposed student branches at Rice Institute and the University of Houston. Reg F. Taylor reported on the progress of the chapter for the first eight months of the fiscal year.

The meeting was turned over to H. L. DeVines who, in the guise of Santa Claus, distributed presents from the grab bag. Attendance 76.

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Virginia—January 13, 1949. Acting Pres. J. E. White welcomed the guests and announced the movie, *Quality Control from Ore to Finished Product*. This 40-minute movie was about seamless copper tubing.

Following, the business meeting was conducted. J. F. Boyenton announced that the speaker at the February meeting would be Al Hanson, vice president, Drayer-Hanson Co. The subject of Mr. Hanson's speech will be *The Heat Pump and Its Application*. E. L. Capps then gave a membership report, which was followed by a report from Sec'y D. E. Phillips. He stated that the chapter had held four meetings during 1948 and the average attendance was 28. During 1948, securing speakers and planning the programs was the responsibility of Mr. Boyenton, program committee chairman.

Mr. Phillips also gave the treasurer's report, after which members voted that all reports be accepted. The meeting was then turned over to Mr. Boyenton, chairman of the nominating committee, and the election of officers was held. The following officers were elected: J. E. White, *president;* W. P. Robinson, *vice president;* D. E. Phillips, *secretary;* W. C. Urrutia, *treasurer;* and R. C. Thomas, J. J. Shanahan, and E. D. Duval, *Board of Governors*. Attendance 37.

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Washington, D. C.—January 12, 1949. The chapter and the Baltimore-Washington section of *ASRE* held their annual joint meeting at the Burlington Hotel, Washington, D.C.

Dinner was preceded by a social hour. Following dinner, the meeting was called to order by W. J. Dugas, a past chairman of the *ASRE* section, who presided as master of ceremonies. A number of impromptu talks were made and entertainment was provided by the *Six Happy Hicks*, a musical organization of the washboard variety. They were roundly applauded.

The master of ceremonies then presented L. Bert Nye, Jr., of the chapter, who introduced the speaker of the evening, Dr. Louis Kayser, dean of students and professor of European history, George Washington University. Dean Kayser, a radio commentator and speaker, spoke on the subject, *The World Today*. He briefly analyzed the points of contention in the Dutch-Indonesian, Arab-Jewish, and Chinese Nationalist-Communist conflicts, and

Annual joint meeting of Washington Chapter ASHVE and Washington-Baltimore Section, *ASRE*, Washington, D. C., January 12, 1949
Left to right at speaker's table: An entertainer; C. W. Hottel, *ASRE*; A. S. Gates, Jr., ASHVE; A. C. Crawford, *ASRE*; L. Bert Nye, Jr., ASHVE; Dr. Louis Kayser; W. J. Dugas, *ASRE*; J. G. Muirheid, ASHVE; P. H. McKim, ASHVE; and W. O. Kline, *ASRE*

JOURNAL SECTION

recommended action to be taken by this country. Attendance 68.

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Western New York—January 10, 1949. Two Society officers, L. E. Seeley, treasurer, and A. V. Hutchinson, secretary, were present at the January meeting which began with dinner at the University Club. Plans for the February meeting were outlined by F. J. Weber who said that this meeting would be held at the New York State Institute of Applied Arts and Sciences, and the program would include a panel discussion with students of the Institute.

ASHVE Secretary Hutchinson reported on the activities of the Society during the previous year and discussed plans for the coming year. On behalf of the chapter, Joseph Davis presented Mr. Hutchinson with a gift consisting of a metal desk set. The set was of modernistic design and contained a letter opener and a pair of scissors. On the cover of the case was a plaque bearing an engraved inscription. The chapter made the gift of this case to the Society's Secretary in appreciation of his long and consistent service with the Society.

The activities of ASHVE were described by Dean Seeley. He stressed the way in which the Society's income is spent, and further reported that student membership is being increased, with no age limit because many of the older boys were coming to school under the G.I. Bill of Rights. The unanimous opinion of the Chapter was that Dean Seeley's talk was very enlightening, giving an insight into the activities of the Society. Pres. Edwin Woolcott presided. Attendance 48.

Western New York—December 15, 1948. The December meeting at Buffalo was an enjoyable one held in the traditional festive spirit of the Christmas season. D. J. Mahoney acted as master of ceremonies. After bowling, cards, and other games, awards were distributed and a buffet supper was served. Attendance 75.

Student Branch

Texas A. & M. College—December 7, 1948. The December meeting in the Fin and Feather Club of Bryan was the annual joint meeting of the Student Branch with the Texas and the Shreveport, La., Chapters. Visitors included: B. S. Foss, Jr., president, North Texas Chapter; C. C. Quin, president, South Texas Chapter; I. W. Wilke, secretary, Southwest Texas Chapter; W. E. Fitz-Gerald, president, Shreveport, La., Chapter; Karl Martino, president, University of Texas Student Branch; and W. E. Long, professor of mechanical engineering, University of Texas. All of these visitors gave short talks.

Guest speaker was John James, chief research engineer for McDonnell & Miller, Inc., Chicago. Mr. James, who was introduced by Julius Blum, outlined the history and development of hot water heating and illustrated his talk with slides of numerous systems and their control. At the conclusion of his talk, a short question and answer period was conducted, with emphasis on control valves.

Pres. S. E. Ammons presided over the meeting. Attendance 97.

C. V. Franks

John Palasics

NEW LABORATORY STAFF MEMBERS

Three engineers have been added to the ASHVE Research Laboratory in Cleveland, according to a recent announcement by L. N. Hunter, chairman, Committee on Research. The new members are: Clifford V. Franks, John Palasics, and Lester Schutrum.

Now serving as a research engineer on the program on panel heating and cooling, Mr. Franks has had 10 years of research experience with the Brush Development Co. and with Battelle Memorial Institute. His experience includes work on underwater detection equipment and on special instruments for measuring surface finish and vibration. In addition, he has investigated magnetic recording devices and has done research on fuels. He has a B.S. degree in physics from Case Institute of Technology.

Mr. Palasics is investigating the characteristics of ventilation jets. His B.S. in mechanical engineering was earned at Fenn College, and he has also taken advanced studies in fluid mechanics and aerodynamics at Case Institute of Technology. Before joining the Laboratory staff, he was research engineer at the Lewis Memorial Propulsion Laboratories of the National Advisory Committee on Aeronautics. There he investigated superchargers and jet propulsion equipment.

Mr. Schutrum's work at the Laboratory is in the capacity of assistant research engineer on the panel heating and cooling research project. A veteran of World War II, he was three years in the U. S. Army Signal Corps, serving with the Radio Intelligence Division, with 18 months in the ETO. He holds a B.S. in Mechanical Engineering from Fenn College.

JOURNAL SECTION

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When the replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month, 234 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by March 25, 1949, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

ABERDEEN, E. A., Sales Repr., John Inglis Co., Ltd. PROPOSERS: J. J. Lawlor, W. R. Blackhall. SECONDERS: R. A. Leith, V. J. Jenkinson.

BANKS, J. B., Regional Mgr., Minneapolis-Honeywell Regulator Co. (*Advancement*). PROPOSERS: J. T. Burtchaell, R. E. Chase. SECONDERS: S. D. Peterson, William Gehrs.

BARRETT, D. M., Avery Engineering Co. PROPOSERS: Dennis Boggs, Lester T. Avery. SECONDERS: L. C. Burkes, H. R. Weiner.

BECKER, H. P., Chief Engr., Center Air Conditioning Corp. PROPOSERS: P. B. Gordon, Shigeki Hiratsuka. SECONDERS: A. A. Bearman, Bernard Leventhal.

BENDIX, C. B. B., Automatic Burner Corp. PROPOSERS: R. F. Connell, A. A. Emrick. SECONDERS: R. E. Moore, C. E. Lewis.

BENZ, D. L., Sales Engr., The Trane Co. PROPOSERS: E. R. Lokey, R. C. Chewning. SECONDERS: W. D. Maxwell, J. P. McDermott.

BEQUILLARD, M. E., Owner. PROPOSERS: J. R. Howard, W. G. Lathrop. SECONDERS: G. F. Murray, J. B. Landry.

BOHREN, A. H., Partner, Bohren Plumbing Service & Supply Co. (*Advancement*). PROPOSERS: A. B. Morrison, Dick Blankenship. SECONDERS: K. H. Hanson, A. N. Hoss.

BRAAK, J. W., Sales Engr., Johnson Service Co. PROPOSERS: A. W. Cooper, P. C. von Rosenberg. SECONDERS: H. H. Herman, F. C. Allen.

BRIDGE, T. E., Design Engr., D. B. Wesstrom. PROPOSERS: A. S. Robinson, M. G. Kershaw. SECONDERS: John Campbell*, G. L. Robinson.

BROWN, C. L., JR., Student-Undergraduate Asst., University of Illinois. PROPOSERS: S. Konzo, R. J. Martin*. SECONDERS: J. R. Carroll, Jr., J. R. Fellows.

BULLER, E. L., Sales Engr., Perkins LeNoir Co. PROPOSERS: E. S. Wilkins, C. A. Corbit. SECONDERS: A. C. Perkins*, H. R. Keperling.

BURG, I. X., Advertising & Sales Promotion Mgr., Utility Appliance Corp. PROPOSERS: J. B. Venneman, J. S. Earhart. SECONDERS: R. J. Petersen, Leo Hungerford.

BURLA, H. W., Swiss Repr. Sales Mgr., Svenska - Flaktfabriken. PROPOSERS: Walter Hausler, L. F. de Bruyn. SECONDERS: R. H. Goerg, Aldo Gini.

BURTCHAELL, J. T., Pres.., Rushlight's Inc. (*Advancement*). PROPOSERS: J. D. Kroeker, B. W. Farnes. SECONDERS: J. B. Banks, S. D. Peterson.

BYRD, B. L., Engr., Paul Scheurer Engineering Co. PROPOSERS: J. D. Poythress, J. O. Gillham*, F. L. Gray.

CARLSON, R. L., Sales Engr., Arthur Forsyth Co. PROPOSERS: K. R. Murhard, L. R. Jones. SECONDERS: C. W. Brissenden, J. B. Banks.

CLARK, JAMES E., Chief Fan Engr., General Blower Co. PROPOSERS: G. F. Spooner, M. M. Hattis. SECONDERS: G. D. Haberer, L. L. Solstad.

CLARK, M. V., JR., Air Cond. & Htg. Design Engr., United Electric Service. PROPOSERS: M. E. Phillips, O. H. Atkins. SECONDERS: W. E. FitzGerald, A. H. Otto.

CLAYTON, W. N., Student, Purdue University. PROPOSERS: W. T. Miller, F. B. Morse. SECONDERS: L. M. Zoss, D. M. Long.

COLLINS, W. J., JR., Partner, Collins & Gould, Cons. Engr. PROPOSERS: J. R. Patten, G. T. Donceel. SECONDERS: H. S. Shafer, J. H. Carnahan.

CONNOR, T. C., Asst. Mech. Engr., Panama Railroad Co., Commissary Div. PROPOSERS: Prof. A. A. Locke*, Prof. G. H. Tweney*. SECONDERS: L. M. Patrick*, Dr. G. P. Loweke*.

COOKE, E. R., Asst. Mgr., Cooper Supply Co. PROPOSERS: A. D. Holmes, Edwin Jones. SECONDERS: H. W. Looney, R. E. Pauling.

CRAIG, J. W., Design-Engr., C. F. Braun & Co. PROPOSERS: Nicholas Nassir, Leo Hungerford. SECONDERS: M. C. Greiner, A. J. Hess.

CULBERTSON, REAVER, Mgr. & Head of Co., B. E. Culbertson Co. PROPOSERS: E. T. Houlihan, J. R. Murphy. SECONDERS: H. A. Essley, R. T. Schoerner.

DEVERO, L. T., Mgr., Armstrong Cork Co. PROPOSERS: Frank McPherson*, W. R. Lee. SECONDERS: Boyd Purifoy, R. F. Shoemaker.

DEVORE, A. B., Sales, American Radiator & Standard Sanitary Corp. PROPOSERS: E. H. Lloyd, W. E. Kingswell. SECONDERS: W. D. Marshall, R. A. Grant.

DOERING, T. R., Sales Engr., Waterman & Waterbury Co. PROPOSERS: A. B. Algren, W. A. Swenberg. SECONDERS: D. E. Sedgwick, Leland Lincoln.

DUKE, R. E., Norman S. Wright & Co. PROPOSERS: N. S. Wright, Jr., W. L. Johnson. SECONDERS: E. R. Lokey, K. R. Murhard.

DWYER, AUSTIN, Heating Field Office Mgr., J. L. Murphy, Inc. PROPOSERS: R. R. Ferguson, William Drew. SECONDERS: Arthur Ritter, M. C. Giannini.

DYSON, R. H., Vice Pres., Canadian Ice Machine Co., Ltd. PROPOSERS: H. R. Roth, M. W. Shears. SECONDERS: R. A. Stencel, R. S. Mathison.

EILBAUM, LOUIS, Pres., Aircon Heating Co., Inc. PROPOSERS: Louis Rubin, C. F. Mally. SECONDERS: T. H. Mabley, M. H. Gross.

ELIZARDI, RALPH, Air Cond. Engr., Cary B. Gamble & Associates. (*Advancement*). PROPOSERS: T. A. Stokes, C. B. Gamble. SECONDERS: G. E. May, W. H. Grant, Jr.

* Non-member

JOURNAL SECTION

ESTES, EDWIN C., Chief Draftsman, Northern Pacific Railway. (*Advancement*). PROPOSERS: A. B. Algren, F. A. Morris. SECONDERS: E. A. Johnson, R. E. Backstrom.

EWINGER, J. W., Htg. Engr., Ewinger Supply Co. PROPOSERS: Milton Meuler*, H. W. Leake*. SECONDERS: P. H. Bacher*, W. C. White*.

FARMAN-FARMAIAN, FAROUGH, Student, University of Illinois. PROPOSERS: J. R. Carroll, Jr., S. Konzo. SECONDERS: J. R. Fellows, A. N. Fleming.

FINKBEINER, J. E., Mfrs. Repr., E. A. Finkbeiner. PROPOSERS: W. B. Morrison, Dick Blankenship. SECONDERS: K. H. Hanson, C. W. Brissenden.

FRAZIER, J. E., Pres. & Secy., Frazier-Simplex, Inc. (*Advancement*). PROPOSERS: J. F. Collins, Jr., R. A. Miller. SECONDERS: E. H. Hobbie, J. S. Herbert.

FRENGER, GUNNAR, Tech. Mgr., C. M. Mathiesen & Co. PROPOSERS: G. L. Wiggs, D. L. Lindsay. SECONDERS: N. Alfsen, Alf Tjersland.

FURST, R. B., Design Engr., C. F. Braun & Co. PROPOSERS: Nicholas Nassir, Leo Hungerford. SECONDERS: M. C. Greiner, A. J. Hess.

GESELBRACHT, W. G., Design Engr., C. F. Braun & Co. PROPOSERS: Nicholas Nassir, Leo Hungerford. SECONDERS: M. C. Greiner, A. J. Hess.

GILHAM, W. C., Student, Purdue University. PROPOSERS: W. T. Miller, F. B. Morse. SECONDERS: L. M. Zoss, D. M. Long.

GRANEK, GERHARD, Htg. & Vtg. Engr., Hydro-Electric Power Commission of Ontario. PROPOSERS: F. G. Ewens, J. R. Petrinec. SECONDERS: John Thompson, D. G. Huber.

HACKL, A. J., Mgr., The Trane Co. PROPOSERS: G. S. Flinn, E. G. Woodson. SECONDERS: L. S. Weil, G. C. Burr.

HARRIS, H. A., Design Engr., C. F. Braun & Co. PROPOSERS: Nicholas Nassir, Leo Hungerford. SECONDERS: M. C. Greiner, A. J. Hess.

HERNDON, C. L., Sales Engr., Straus-Frank Co. PROPOSERS: L. S. Pawkett, L. H. Hornoer, Jr. SECONDERS: K. A. Monier, I. W. Wilke.

HERRICK, H. M., Air Cond. Designer, C. F. Braun & Co. PROPOSERS: Nicholas Nassir, Leo Hungerford. SECONDERS: M. C. Greiner, A. J. Hess.

HOLLAND, A. W., Chief Engr., Refrigeration Equipment Co., Inc. PROPOSERS: L. V. Busenlener, E. O. Thoman. SECONDERS: C. V. Bankston, W. H. Grant, Jr.

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SHERMAN RECEIVES PERCY NICHOLLS AWARD

The 1948 Percy Nicholls award was recently given to Ralph A. Sherman, ASHVE member, for achievement in research on solid fuels. The award was made by the *ASME* and *AIME*.

R. A. Sherman

Mr. Sherman is assistant director at Battelle Memorial Institute, Columbus, Ohio, and has been a member of ASHVE since 1933. Since 1946, he has been on the Society Council.

PERSONALS

John C. Barnes is now vice president for sales in the eastern division of Rittling Corp., Buffalo, N. Y. Mr. Barnes was formerly with the National Radiator Co., Philadelphia. He is a member of ASHVE.

Earl L. Bedard, ASHVE member, has been elected vice president of Mt. Hawley Mfg. Co., Peoria, Ill. In addition to his new duties, Mr. Bedard will continue as sales manager, a position he has held since the beginning of 1948.

Four promotions in the departments of research and engineering have been announced by the Philco Co. Among the men named to head principal appliance design groups is Matson C. Terry, who is now chief engineer in charge of developing single-room conditioners. He has been a member of ASHVE since 1936.

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UTAH	R. H. East	G. C. Campbell
VIRGINIA	D. E. Phillips	J. M. Stokley
WASHINGTON, D. C.	A. S. Gates, Jr.	P. R. Achenbach
WESTERN MICHIGAN	C. H. Pesterfield	T. D. Stafford
WESTERN NEW YORK	Roswell Farnham	F. J. Weber
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6th Oregon State A. C. Conference

A. E. Stacey, Jr., president of ASHVE, will be a featured speaker, and a number of technical papers will be presented at the Oregon State Air Conditioning Conference to be held at Corvallis, Ore., March 24 to 26.

Sponsored by the Oregon Chapter, ASHVE, and the mechanical engineering department of Oregon State College, the 6th Air Conditioning Conference will be a 3-day affair, with 5 technical sessions and a banquet on the schedule.

March 24—Morning, Registration. Afternoon, Technical Sessions. Evening, Technical Sessions open to Public.

March 25—Morning, Technical Sessions. Afternoon, Technical Sessions. Evening, Banquet & Annual Meeting.

March 26—Morning, Technical Sessions.

Kenneth N. Flocke is chairman of the General Committee. Chairmen of other committees are: John B. Banks, annual meeting and banquet; Earl C. Willey, arrangements; Albert C. Clark, attendance; Richard P. Lankow, publicity; and Robert W. Martig, technical program.

5th Texas A. C. Conference

Four ASHVE chapters are participating in the 5th Annual Air Conditioning Conference at A & M College of Texas, College Station, March 28, 29, and 30.

Executive chairman of the Conference is Bert P. Fisher, Houston, with A. J. Rummel, San Antonio, Marvin Brown, Dallas, and William E. FitzGerald, Shreveport, La., as co-chairmen.

The Conference will be sponsored by the North Texas, South Texas and Southwest Texas Chapters and the Shreveport Chapter, in conjunction with the college.

Engineers' Council of Houston

Five speakers including Louis Bromfield will take part in the Second Annual Symposium of the Engineers' Council of Houston at the Rice Hotel, Houston, Tex., on April 2.

The symposium is being built around the theme, *Conservation of Our Natural Resources.*

Reservations may be made through Dean F. Saurenman, P. O. Box 3048, Houston 1, Tex.

JOURNAL SECTION

OFFICERS OF LOCAL CHAPTERS AND STUDENT BRANCHES

(41 Chapters—3 Student Branches)

Atlanta: Organized, 1937. Headquarters, Atlanta, Ga. Meets, First Friday. President, Leo Sudderth, Jr., 810 Bona Allen Bldg., Atlanta 3. Secretary, F. A. Player, 313 Techwood Dr., N. W., Atlanta.

•

Central New York: Organized 1944. Headquarters, Syracuse, N. Y. President, D. L. Mills, 1407 N. Madison St., Rome. Secretary, J. H. Carpenter, 3515 Midland Ave., Syracuse 7.

Central Ohio: Organized, 1944. Headquarters, Columbus, Ohio. Meets, Third Monday. President, W. M. Myler, Jr., P.O. Box 267, Columbus 16. Secretary, H. G. Hays, 851 W. Third Ave., Columbus 8.

Cincinnati: Organized 1932. Headquarters, Cincinnati, Ohio. Meets First Tuesday. President, M. E. Mathewson, 1211 Enquirer Bldg., Cincinnati 2. Secretary, E. W. McNamee, 1729 John St., Cincinnati 14.

Connecticut: Organized, 1940. Headquarters, New Haven, Conn. Meets, Third Thursday. President, E. C. Marsden, 44 Hicks St., Hartford. Secretary, T. L. Arnold, 26 Francis Ave., Hartford.

•

Delta: Organized, 1939. Headquarters, New Orleans, La. Meets, Second Tuesday. President, L. V. Busenlener, 2927 Jackson Ave., New Orleans 13. Secretary, Ralph Elizardi, 314½ St. Charles Ave., New Orleans 12.

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Golden Gate: Organized, 1937. Headquarters, San Francisco, Calif. Meets, First Thursday. President H. V. Hickman, 1129 Folsom St., San Francisco 1. Secretary, R. C. Cushing, 1136 Howard St., San Francisco 3.

•

Illinois: Organized, 1906. Headquarters, Chicago, Ill. Meets, Second Monday. President, C. M. Burnam, Jr., 6 N. Michigan Ave., Chicago 2. Secretary, G. W. Bornquist, 629 W. Washington Blvd., Chicago 6.

Indiana: Organized, 1943. Headquarters, Indianapolis, Ind. Meets, Fourth Friday. President, P. R. Jordan, 311 E. South St., Indianapolis 4. Secretary, C. F. A. Locke, 111 E. 34th St., Indianapolis 5.

Iowa: Organized 1940. Headquarters, Des Moines, Ia. Meets, Second Tuesday. President, D. C. Murphy, 840 Fifth Ave., Des Moines 14. Secretary, R. S. Stover, Kresge Bldg., Marshalltown.

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Kansas City: Organized, 1917. Headquarters, Kansas City, Mo. Meets First Monday. President, H. E. Gould, 1924 Oak Rd., Kansas City 8. Secretary, K. M. Stevens, 1510 Main St., Kansas City.

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Manitoba: Organized, 1935. Headquarters, Winnipeg, Man., Canada. Meets, Third Thursday. President, J. R. Stephenson, 838 Notre Dame Ave., Winnipeg. Secretary, D. F. Michie, 838 Notre Dame Ave., Winnipeg.

Massachusetts: Organized, 1912. Headquarters, Boston, Mass. Meets, Third Tuesday. President, R. T. Kern, 51 Claflin St., Leominster. Secretary, D. W. Blair, 87 Withington Rd., Newtonville.

Memphis: Organized, 1944. Headquarters, Memphis, Tenn. Meets, Third Monday. President, E. E. Scott, 918 Dermon Bldg., Memphis 3. Secretary, A. T. Bevil, 1521 Waverly Ave., Memphis 6.

Michigan: Organized, 1916. Headquarters, Detroit, Mich. Meets, First Monday after 10th of month. President, E. F. Hyde, 708 Oakland Ave., Birmingham. Secretary, J. H. Spurgeon, 5050 Joy Rd., Detroit 4.

Minnesota: Organized, 1918. Headquarters, Minneapolis, Minn. Meets, First Monday. President, O. L. Lilja, 5000 16th Ave., S., Minneapolis 7. Secretary, E. F. Snyder, Jr., 4324 Zenith Ave., N., Minneapolis 12.

Montreal: Organized, 1936. Headquarters, Montreal, Que., Canada. Meets, Third Monday. President, Leo Garneau, 25 Salisbury Rd., Pointe Claire, Que. Secretary, R. R. Noyes, 630 Dorchester St. W., Montreal.

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Nebraska: Organized, 1940. Headquarters, Omaha, Neb. Meets, Second Tuesday. President, K. R. Magarrell, 127 So. Main St., Council Bluffs, Ia. Secretary, F. E. Prawl, 2820 Harney St., Omaha 2.

New York: Organized, 1911. Headquarters, New York, N. Y. Meets, Third Monday. President, A. A. Bearman, 444 West 56th St., New York 19. Secretary, Carl H. Flink, Room 3000, 51 Madison Ave., New York 10.

North Carolina: Organized, 1939. Headquarters, Durham, N. C. Meets, Quarterly. President, DeParx Stimson, 121 Idlewilde Dr., Winston-Salem. Secretary, R. B. Crosland, Jr., 1001 S. Marshall St., Winston-Salem.

North Texas: Organized 1938. Headquarters, Dallas, Tex. Meets, Third Monday. President, G. A. Linskie, Buckner Blvd., Dallas 10. Secretary, A. B. Ulrich, Jr., 715 Lipscomb St., Dallas 14.

Northeastern Oklahoma: Organized, 1948. Headquarters, Tulsa, Okla. President, W. R. Lee, 303 Beacon Bldg., Tulsa 3. Secretary, R. F. Shoemaker, 1136 So. Peoria, Tulsa.

Northern Ohio: Organized, 1916. Headquarters, Cleveland, Ohio. Meets, Second Monday. President, W. R. Moore, 5005 Euclid Ave., Cleveland. Secretary, John Richmond, 10210 Woodland Ave., Cleveland 4.

Oklahoma: Organized, 1935. Headquarters, Oklahoma City, Okla. Meets, Second Monday. President, J. H. Spaan, Jr., 1303 N. Klein, Oklahoma City 6. Secretary, J. R. Patten, 627 N.W. Second St., Oklahoma City.

Ontario: Organized, 1922. Headquarters, Toronto, Ont., Canada. Meets, First Monday. President, E. G. Spall, 73 Simcoe St., Toronto 1. Secretary, H. R. Roth, 57 Bloor St., W., Toronto, Ont.

Oregon: Organized, 1939. Headquarters, Portland, Ore. Meets, Thursday after First Tuesday. President, E. R. Lokey, 10340 N.E. Sandy Blvd., Portland 13. Secretary, A. N. Hoss, 621 S. W. Alder, Portland.

•

Pacific Northwest: Organized, 1928. Headquarters, Seattle, Wash. Meets, Second Tuesday. President, L. L. Bysom, 1214 Eighth St., Bremerton. Secretary, D. O. Mead, 816 Arctic Bldg., Seattle 4.

Philadelphia: Organized, 1916 Headquarters, Philadelphia, Pa. Meets, Second Thursday. President, J. O. Kirkbride, Fourth & Locust Sts., Philadelphia 6. Secretary, M. E. Barnard, 12 S. 12th St., Philadelphia 7.

Pittsburgh: Organized, 1919. Headquarters, Pittsburgh, Pa. Meets, Second Monday. President, H. E. Park, 201 E. Carson St., Pittsburgh 19. Secretary, E. H. Riesmeyer, Jr., 231-33 Water St., Pittsburgh 22.

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Rocky Mountain: Organized, 1944. Headquarters, Denver, Colo. Meets, First Wednesday. President, Fred Janssen, 1123 W. Third Ave., Denver 9. Secretary, F. C. Allen, 4467 King St., Denver 11.

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St. Louis: Organized, 1918. Headquarters, St. Louis, Mo. Meets, First Tuesday. President, B. L. Evans, 371 Stratford Ave., University City. Secretary, C. H. Burnap, 1635 Syndicate Trust Bldg., St. Louis 1.

Shreveport: Organized 1948. Headquarters, Shreveport, La. Meets, Third Thursday. President, A. H. Otto, 1020 Market St., Shreveport. Secretary, S. W. Beaty, P. O. Box 1734, Shreveport.

South Texas: Organized, 1938. Headquarters, Houston, Tex. Meets, Third Friday. President, C. C. Quin, 809 Stuart Ave., Houston 6. Secretary, R. J. Salinger, 4510 Dewberry Lane, Houston 4.

Southern California: Organized, 1930. Headquarters, Los Angeles, Calif. Meets, Third Wednesday. President, R. S. Farr, 2615 Southwest Dr., Los Angeles. Secretary, L. J. Helms, 633 S. Berendo St., Los Angeles 5.

Southwest Texas: Organized, 1946. Headquarters, San Antonio, Tex. Meets, Second Tuesday. President, G. R. Rhine, 1206 Alamo National Bldg., San Antonio. Secretary, I. W. Wilke, 422 Congress Ave., Austin.

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Utah: Organized, 1944. Headquarters, Salt Lake City, Utah. Meets, First Wednesday. President, E. J. Watts, 2485 Douglas St., Salt Lake City 5. Secretary, R. H. East, 326 E. First So., Salt Lake City.

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Virginia: Organized, 1946. Headquarters, Norfolk, Va. President, J. E. White, 7607 Gloucester Ave., Norfolk 5. Secretary, D. E. Phillips, P. O. Box 674, Newport News.

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Washington, D. C.: Organized, 1935. Headquarters, Washington, D. C. Meets, Second Wednesday. President, A. S. Gates, Jr., 621 N. Jackson St., Arlington, Va. Secretary, J. G. Muirheid, 2013 New Hampshire Ave., N. W., Washington 9.

Western Michigan: Organized, 1931. Headquarters, Grand Rapids, Mich. Meets, Second Monday. President, H. W. Wolters, 820 Staples Ave., N.W., Kalamazoo 54. Secretary, H. R. Limbacher, 760 E. Vine St., Kalamazoo.

Western New York: Organized, 1919. Headquarters, Buffalo, N. Y. Meets, Second Monday. President, Edwin Woolcock, 2217 15th St., Niagara Falls, N. Y. Secretary, T. F. Killeen, 374 Delaware Ave., Buffalo 2.

Wisconsin: Organized, 1922. Headquarters, Milwaukee, Wis. Meets, Third Monday. President, W. A. Ouweneel, 514 E. Ogden Ave., Milwaukee 2. Secretary, H. F. Brinen, 2226 Kinzie Ave., Racine.

Student Branches

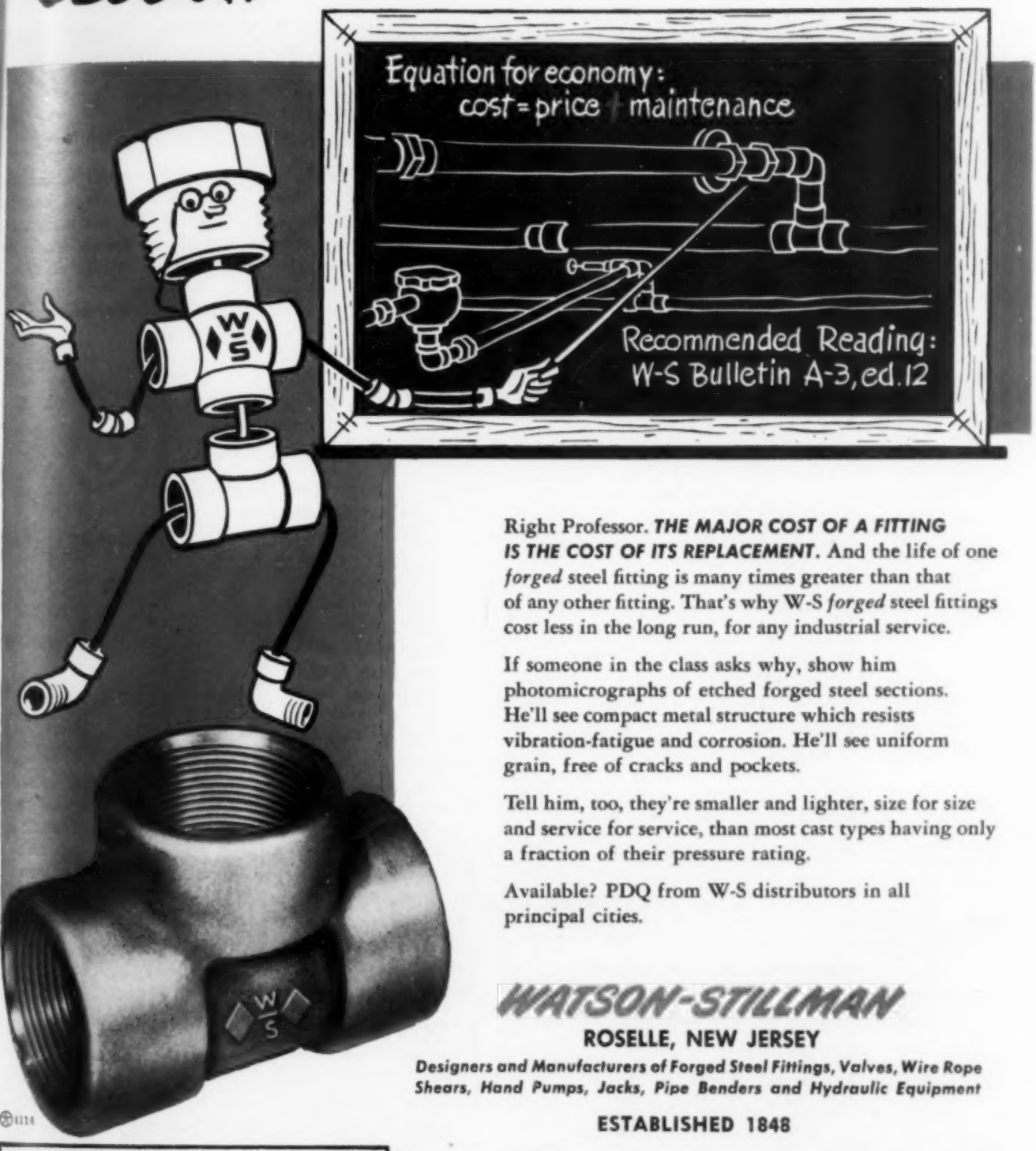
North Carolina State College: Organized 1948. Headquarters, Raleigh, N. C. President, J. G. Marshall, North Carolina State College. Secretary, Basil Greene, Apt. D-2, Grosvenor Garden Apts., Raleigh.

Purdue University: Organized 1948. Headquarters, W. Lafayette, Ind. Chairman, M. E. Naftzger, FPHA 228-2, State St., W. Lafayette. Secretary, C. M. Wade, The Maples Trailer Camp, Lafayette.

Texas A. & M. College: Organized, 1946. Headquarters, College Station, Tex. Meets, Third Thursday. President, C. V. Brown, Mech. Engrg. Dept., Texas A. & M. College, College Station. Secretary, R. N. Hughes, Box 914, College Station.

A FITTING LESSON

FORGED FITTINGS LAST



Right Professor. ***THE MAJOR COST OF A FITTING IS THE COST OF ITS REPLACEMENT.*** And the life of one *forged* steel fitting is many times greater than that of any other fitting. That's why W-S *forged* steel fittings cost less in the long run, for any industrial service.

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ALLOYS
Stainless types 304—316—347
Chrome Moly—Carbon Moly—
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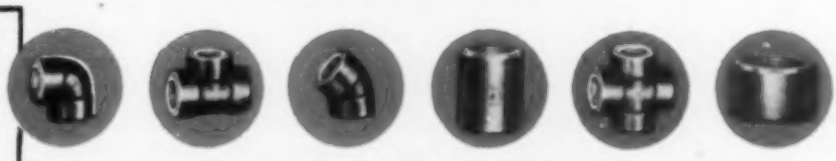
Special alloys for Low Temperatures



SOLD THROUGH LEADING DISTRIBUTORS

EQUIPMENT DEVELOPMENTS

For your convenience in obtaining more information about any of this equipment, see coupon on this page. Add the new products and companies listed here to your Directory Section which you received in your January 1949 Heating, Piping & Air Conditioning and thus keep your records of sources of supply up to date throughout the year. Single asterisk indicates equipment not listed in Directory Section; double asterisk equipment and manufacturer not listed.

Continuous U Bend Copper Coils Used in New Line of Unit Heaters

No. 3985—A new line of unit heaters, model 46U, for either steam or hot water has been brought out by Carrier Corp., 302 S. Geddes St., Syracuse 1, N. Y.

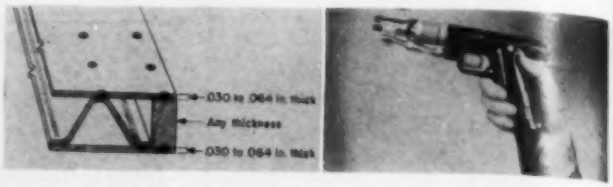
The most outstanding feature, says the company, is the use of one row "Aerofin" coils of continuous U bend copper tubing brazed into steel headers. Deep and wide fin spacing is said to minimize resistance to air flow, to facilitate cleaning, and to provide better heat transfer.

Streamlined in style and light in weight, units are designed for ease of installation and servicing and are said to be unusually quiet in operation.

Spot Welding Torch Provides Inert Gas-Shielded Arc

No. 3986—A portable, inert gas shielded spot welding torch has been developed by Linde Air Products Co., Unit of Union Carbide and Carbon Corp., 30 E. 42nd St., New York 17. Called the "Heliarc" HW-8, it is designed to make spot welds from only one side of the work without the necessity of backing the weld location. The sketch shows a typical use in the construction of a corrugated sandwich.

Mild steel, low alloy, and stainless steel, 0.030 to 0.06[illegible] in. thick can be welded using currents from 15[illegible] to 25[illegible] amp. Not only can two sheets of these metals be joined in 1 to 2 sec per spot, but also a sheet of metal can be joined to underlying material of any thickness. This makes it possible to spot weld corrosion resistant sheets to mild steel so as to provide cladding—says the manufacturer.



The torch is said to be especially useful where the structure is large or of a complicated shape because welding is done from one side and no forging pressure is required. Argon gas is used to shield the electrode and weld area to prevent oxidation. Welding duration and argon flow are controlled by a separate timer unit. This torch uses a-c or d-c with high frequency starting and stabilization of the arc.

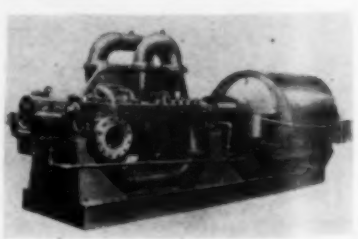
The standard hose assembly, 25 ft long, is about an inch in diameter and contains conductors for power cable, cooling water, shielding gas, and trigger control. Electrode adjustment is made by using a push button release located on the torch.

Many Improvements Feature Evaporative Condensers

No. 3987—Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, has placed on the market a line of evaporative condensers, in a range of sizes from 3 to 50 tons (duplexed to 100 tons).

Featured improvements in the line are said to include: (1) "Penta-Post" frame members so designed as to make possible multiple service and access panels in the welded frame assembly, (2) all prime surface coils with copper tubes hydraulically expanded at 3000 psi into copper tube sheets with ferrule collars, (3) for ammonia, hot dipped galvanized steel coil assemblies, (4) hot dipped galvanized sump fan, blower and coil frame sections, wheels, and scrolls, (5) copper spray headers, and (6) heavy steel coil side frames, galvanized after fabrication.

Adds Four Boiler Fuel Pumps to Line of Centrifugal Units



No. 3988 — Worthington Pump and Machinery Corp., Harrison, N. J., has added four new boiler feed pump sizes to its line of centrifugal pumps.

Two of the new sizes are 6 in. pumps and two are 8 in. pumps.

FOR YOUR CONVENIENCE

Heating, Piping & Air Conditioning [3-49]
6 N. Michigan Ave., Chicago 2, Ill.

Please ask the manufacturers to send me more information about the equipment mentioned under the following reference numbers in Equipment Developments and Recent Trade Literature

(Circle each number in which you are interested)

3985	3986	3987	3988	3989	3990	3991	3992	3993	3994
3995	3996	3997	3998	3999	4000	4001	4002	4003	4004
7165	7166	7167	7168	7169	7170	7171	7172	7173	7174
7175	7176	7177	7178	7179	7180	7181	7182	7183	7184
7185	7186	7187							

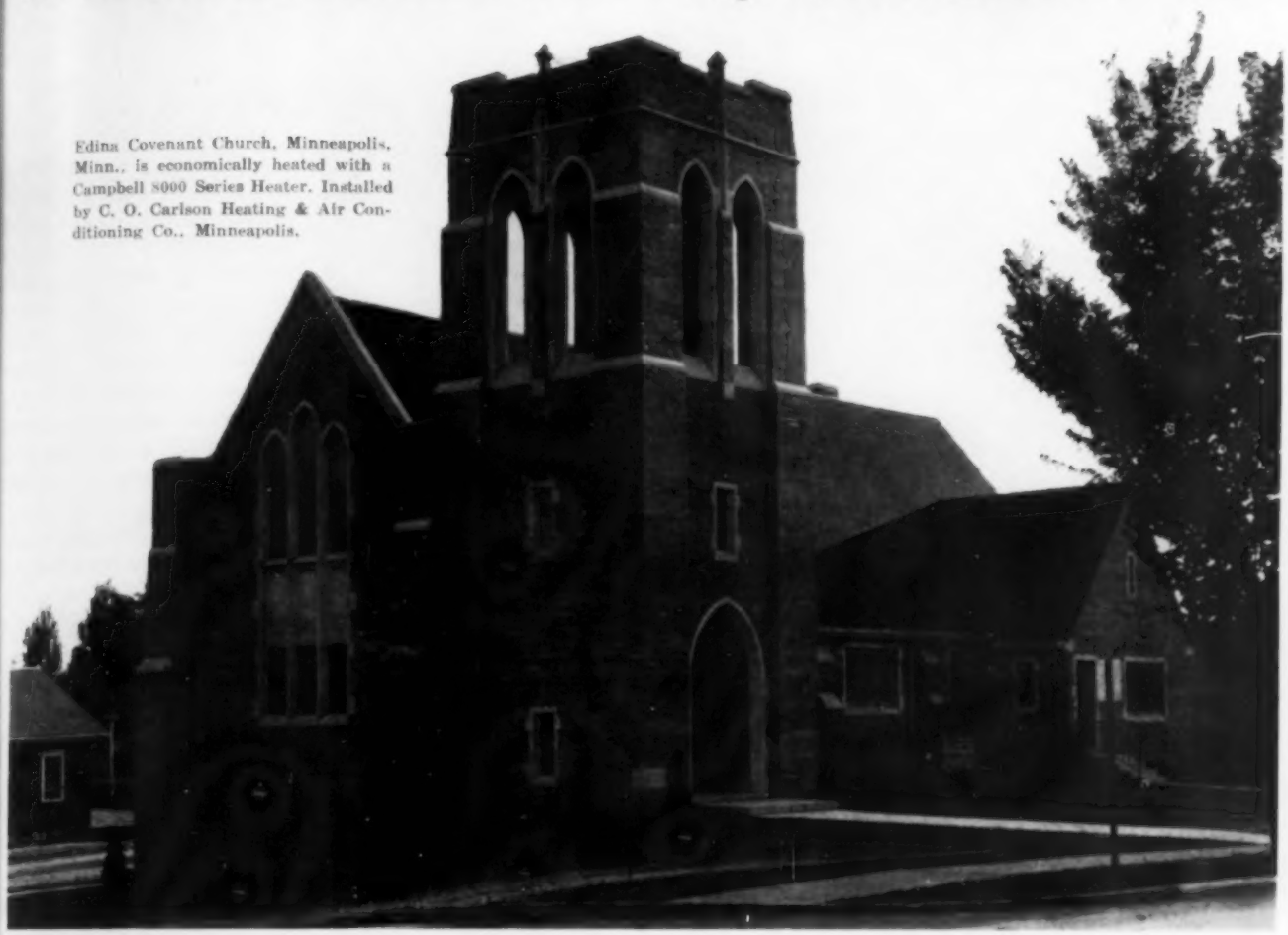
Name ..... Title .....

Company .....

Address .....

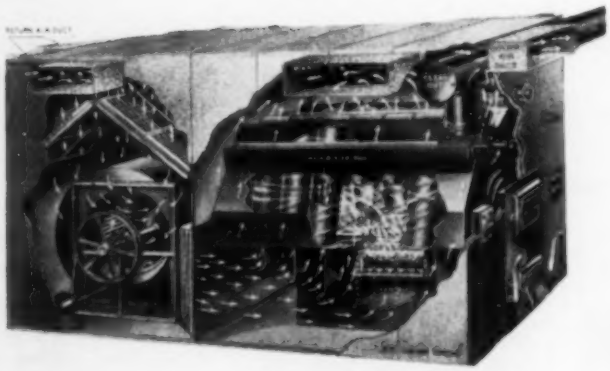
City ..... Zone ..... State .....

WARM AIR HEATED With Only 4,000 Gallons of Oil



Edina Covenant Church, Minneapolis, Minn., is economically heated with a Campbell 8000 Series Heater. Installed by C. O. Carlson Heating & Air Conditioning Co., Minneapolis.

REMARKABLE fuel economy, plus positive control of temperature, humidity and ventilation, make Campbell 8000 Heaters especially adaptable for churches, schools and other large buildings. For all the advantages of efficient warm air heating, investigate the Campbell 8000 Series for your next installation. Available for all fuels in capacities up to 2,160,000 B.T.U.'s. Our engineering department will be glad to provide layouts and sketches for any large warm air heating installation.



We also build direct oil and gas-fired heaters for drying and processing operations, up to 14,000,000 B.T.U.'s.

Campbell
HEATING COMPANY
Des Moines 17, Iowa

Every Campbell Heater backed by over 65 years of Heating Experience

You ALL do better with ALDRICH

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Investigate the EXTRA ADVANTAGES you get with Aldrich! Six sizes of boiler-burner units with capacities of 118,000 to 808,000 BTU per hour for low-pressure steam or hot water systems . . . 6 sizes for hot water supply with ranges of 125 to 850 GPH at 100 degrees rise. Each has identical ratings with *interchangeable oil and gas burners*. Many other features to assure low cost installation, maintenance, operation!

Aldrich Oil Burners, with Aldrich-built transformers and motors and complete automatic controls, come in 3 models, 5 firing ranges, ¾ GPH to 19 GPH. Exclusive design and construction features make them ideal for conversion, replacement, or original equipment. Write for literature! Get the EXTRA ADVANTAGES only Aldrich offers!

JOBBERS
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...with ALDRICH

you've got what they want!

You'll find that your sales pick up immediately when you put the same effort into selling Aldrich that you put into another line . . . for Aldrich has a wider variety of sizes, EXTRA competitive advantages.

With Aldrich you offer a heating and/or hot water supply unit that meets the exact needs of each prospect.

A big sales clincher for boiler-burner prospects are the interchangeable, identically rated oil and gas burners. Other features include exceptional compactness, improved design for more economical operation with either gas or oil, and such practical advantages as larger access plates for easier servicing.

Aldrich Oil Burner Advantages include interchangeable draft-tube assemblies giving one burner three firing ranges — you can serve a maximum number of prospects with minimum stock. Construction and performance features of the Aldrich let you meet and beat any competition.

Get the facts!

Get profits you may be missing now. Find out how Aldrich lets you offer something extra in efficient heating and hot water supply to residences, business buildings, industrial plants, institutions, dairies, restaurants. Find out how you can be a party to more of those happy business deals where everybody's satisfied!

ALDRICH HEAT-PAK PRODUCTS

ALDRICH COMPANY

105 East Williams Street, Wyoming, Illinois

A Subsidiary of Breeze Corporations, Inc.

ONE OF THE BREEZE MARK PRODUCTS

[*Continued from page 208*]

Exhaust Control, Two Speed Fan Feature Room Conditioner

No. 3995—Carrier Corp., 302 S. Geddes St., Syracuse 1, N. Y., has developed a new console type room air conditioner, the 51H2 De-Luxe, which carries a five year warranty and is offered in two-tone finish to blend with interior decoration schemes.

Of 1 hp size, it replaces the ¾ and 1½ hp models previously offered. New features include a special exhaust control for the quick removal of stale air and smoke and a two speed fan. Said to be exceptionally quiet, the compressor and motor are hermetically sealed and the compressor starts unloaded.

The standard model operates on 230 or 208 volt, single phase, 60 cycle current. Special models are available for use with 115 volt d-c current.

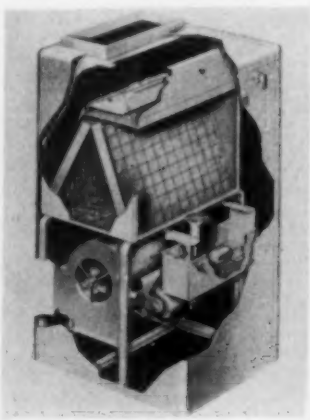
Increases Centrifugal Fan Efficiency

No. 3996—The "Silentvane" centrifugal fan has been redesigned by Sturtevant Div., Westinghouse Electric Corp., 107 Readville Ave., Hyde Park, Boston 36. It is designated the Design 10 and was shown for the first time at the recent 9th International Heating and Ventilating Exposition.

Static efficiency of 80 percent is said to be achieved by a combination of design changes in the blading, wheel inlet, back plate, and scroll. Modification of the scroll so that it is steeper where close to the wheel is said to provide greater uniformity of air distribution, resulting in improved performance and quieter operation.

These fans, with back-tilted blades, are available in 23 sizes and wheel diameters range from 1 to 9 ft.

Uses Triethylene Glycol to Kill Airborne Bacteria



No. 3997—A new self contained, air treating unit for cleaning, freshening, and controlling relative humidity of air is being marketed by National Air Conditioning Inc., Johnstown, Pa. Known as the "TEG" conditioner, the unit vaporizes triethylene glycol into its airstream. This is said to produce a killing action on certain types of airborne bacteria.

The motor driven blower draws air through the unit, over a permanent glass fiber filter cell which is saturated and flushed continuously by

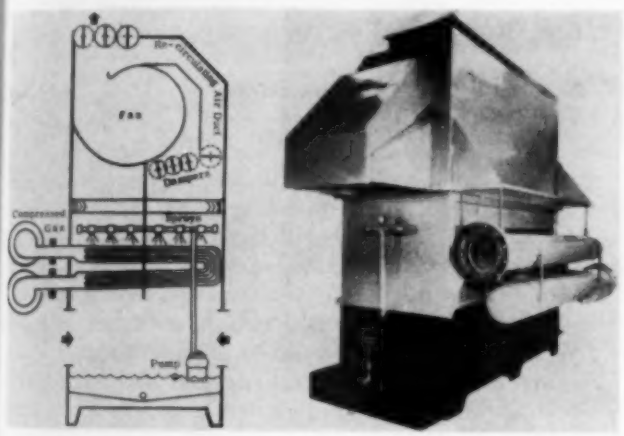
liquid "Tegelene," circulated by a small sump pump.

Air passing through the filter is said to be cleansed of dirt particles by contact with the liquid and the wetted glass fibers, the residue dirt which tends to build up being washed off the cell by the flushing action of the liquid. Humidification is accomplished by the water in the liquid which also picks up a small amount of triethylene glycol vapor. The latter is used to kill certain types of airborne bacteria which cause infectious diseases. This glycol vapor, says the company, is most effective under controlled conditions of relative humidity.

Evaporative Cooler Removes Moisture from Compressed Air, Gas

No. 3998—Shown here is the new model "Aero" after cooler offered by Niagara Blower Co., 405 Lexington Ave., New York 17.

It is designed to remove moisture from compressed air or gas by cooling it to a temperature below the dry bulb temperature of the atmospheric air. This is done by evaporating a recirculating water spray on the surface of a coil through which the compressed air passes, creating a temperature close to the wet bulb temperature of the surrounding atmosphere and lower than the dry bulb temperature and also below the summer surface water temperature.



In the new model, protection against freezing is said to be given by the use of the "balanced wet bulb" control. A thermostat in the spray water reservoir is set for a desired low temperature limit at which it operates dampers to recirculate the air stream internally around a division plate in the center of the unit, instead of drawing in freezing air. This, says the company, permits outdoor installations.

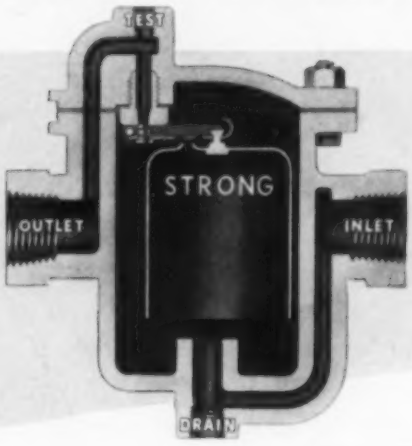
Compactness Features Temperature Controller

No. 3999—Brown Instrument Co., Div. of Minneapolis-Honeywell Regulator Co., 4534 Wayne Ave., Philadelphia 44, has developed a new "low cost" combination motorized valve and temperature controller for numerous industrial processes.

It may be used for on-off, two position, floating, or proportional control. A remote bulb thermal system positions the motorized valve in response to temperature changes, and the unit is available with temperature ranges between -50 and 700 F, and with different sizes and types of valves. The self contained assembly includes valves, linkage, motor, transformer, and temperature controller.

for small drainage jobs . . .

A NEW *SMALL* TRAP WITH *BIG* TRAP QUALITY FEATURES!



- Renewable seat—easily removed for servicing.
- Strong's patented, wear-resistant Anum-Metl† valve and seat—guaranteed leak-proof for one year. Scientifically designed Hi-Cap° orifice gives greater capacity.
- Oversized ports—increased capacity.
- Horizontal inline connections—simplifies piping.
- Stainless 18-8 high ratio leverage.
- Large gasket area.
- Seamless, deep drawn, stainless 18-8 bucket—no welds.
- Bottom drain or inlet connection.

Available for pressures from 0 to 150 p.s.i., the Strong* No. 070 fits all standard trap applications. Built especially for use on laundry, restaurant and hospital equipment and in industry wherever small drainage is a problem.

As with all Strong Semi-Steel Traps, the 070 is guaranteed against defects in material and workmanship for one year. Write today *for more complete information.*

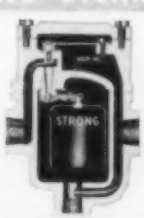
*T. M. Reg. U. S. Pat. Off.

STRONG, CARLISLE & HAMMOND COMPANY
1392 West 3rd Street
Cleveland 13, Ohio

† Anum Metl Reg. Trade Mark

OTHER STRONG STEAM SPECIALTIES

Reducing Valve



Blast Trap

Strainer

STRONG

THESE SCREWS
Answer IMPORTANT QUESTIONS
ABOUT WHICH *Fasteners* TO USE

Uniformity?

Elco heads, threads, slots are made consistently alike under a system of rigid production control. This is assurance against defects and costly delays.

Fast Assembly?

Quick, easy-starting points with accurately formed heads, threads and slots qualify Elco Fasteners for mechanical pick-up and driving. Hours of handling time are saved.

Tensile Strength?

Elco fasteners hold securely in place, continuously under maximum tension, because they are made from special analyzed steels, selected by experience and maintained from constant sources.

Customer Satisfaction?

The dependability of Elco fasteners to provide uniformity, fast assembly and maximum tensile strength means a superior product and trouble-free assembly operation.

Write today and inquire about Elco fasteners to meet your assembly requirements. Send for catalog No. 48.

ELCO Screws

ELCO TOOL AND SCREW CORPORATION
1920 BROADWAY ROCKFORD, ILLINOIS
WOOD SCREWS • MACHINE SCREWS • MACHINE SCREW NUTS
DRIVE SCREWS • CAP SCREWS • LAG SCREWS • SPECIAL SCREWS
TAPPING SCREWS • STOVE BOLTS • PIPE PLUGS

Regulates Speed of Pumps, Other Equipment Electronically

No. 4000—A new low cost "Thy-mo-trol" drive, known as the type H1, is being marketed by General Electric Co., Schenectady 5, N. Y. Furnished in ratings through ½ hp, the drive utilizes a simplified half-wave circuit to provide d-c flexibility from a-c power. It is specially designed to provide, by the turn of a knob at a single control station, stepless speed control for equipment such as pumps, small lathes, grinders, drill presses, conveyors, packaging equipment, and paint and varnish mixers.

The drive operates on 220 volt, 60 cycle a-c (for other voltages a suitable anode transformer is available). It has a 20:1 speed range from 1725 to 86 rpm and operates in ambient temperatures from 50 to 104 F. It is furnished in either the nonreversing or the reversing type, the latter equipped with a separate magnetic reversing switch.

Designed for constant torque loads, under normal conditions the new drive is said to have excellent speed regulation from no load to full load. The speed may be preset or may be varied during operation and under load.

All-Aluminum Construction Used in Electrostatic Air Cleaner

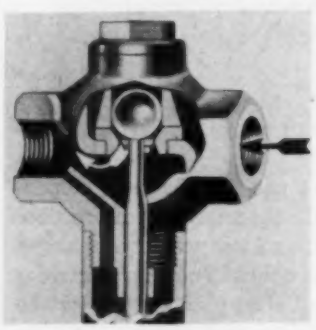
No. 4001—An all-aluminum model of the industrial "Precipitron" that can handle a 10 percent greater volume of air than previous units of the same size has been developed by B. F. Sturtevant Co., Div. of Westinghouse Electric Corp., Hyde Park, Boston 36.

There are 99 plates in the new model, compared to 111 in the old, but 3 in. additional length in each plate is said to give the new unit 17 percent more dirt trapping area.

The new unit will be built in two sizes: 36 in. wide and 24 in. wide. The larger unit is rated at 1000 cfm and the smaller one 667 cfm. Installed in conjunction with ventilation systems, the cells are stacked in tiers according to air cleaning needs.

The 36 in. cell weighs 50 lb, or 65 lb less than the old cell for the same capacity. This lighter weight, combined with improvements in the frame and body construction, is said to make the unit easier and cheaper to install.

Ball Type Inner Valve Used in Pressure Regulating Valves



No. 4002—A ball type inner valve, said to give accurate regulation, great capacity, and tight closing, is now being used on all of the 400 series pressure regulating valves offered by Klipfel Mfg. Co., Div. of Hamilton-Thomas Corp., Hamilton, Ohio.

Designed for applications requiring the accurate regulation of steam, water, air, or

gas at inlet pressures less than 250 psi, reduced pressures below 150 psi, and where reduced pressures must be adjusted frequently, all of the 400 series are single seated with the ball inner valve actuated by a rubber diaphragm. The stainless steel ball is said to center freely on the seat with uniform contact all around. When off the seat, it is free to turn so that all parts of the ball are exposed to the scouring action of the flow.

Redesigned Spray Nozzles Available for Many Uses

No. 4003—"Rotojet" nozzles have been redesigned for a wide range of commercial and industrial applications involving the spraying of brine, oil, water, and other liquids, according to Binks Mfg. Co., 3114-40 Carroll Ave., Chicago 12.

The range of sizes, ⅛ to 2 in., is said to make the nozzles ideal for many uses, from small spray capacity applications to large volume jobs such as in spray ponds, water aerating systems, and cooling towers.

Shown here is a large capacity, heavy duty nozzle of one piece cast brass construction and having an involute type whirl chamber. Smaller units are designed on the side inlet, whirl chamber principle and are of two piece brass construction (though they can be furnished in monel, stainless steel, manganese bronze, or other machinable metals obtainable in bar stock).

Turbine Pump Line Includes Units for Many Services

No. 4004—A new line of medium duty turbine pumps, available in many types from Roy E. Roth Co., Rock Island, Ill., has been designed as a low priced line for use in water supply service, condensate return, boiler feed service, or any application where nonviscous and nonabrasive liquids are handled against moderate pressures.

Capacities are from 2 to 10 gpm at 20 to 100 ft head and at 1750 rpm or 2 to 8 gpm at 100 to 200 ft head and at 3450 rpm.

The only moving part is the impeller, mounted on a stainless steel shaft which turns on two ball bearings. There is only one stuffing box.

Also announced as available from stock are special stainless steel pumps for process work in refineries, chemical plants, synthetic rubber plants, distilleries, food processing plants, and other industries. These have capacities from 2 to 180 gpm at 0 to 400 ft head.

The manufacturer claims that the stainless units can develop high pressures in a single stage and are self venting.

Equipment Shorts

Carrier Corp., Syracuse, N. Y., has released a new 16 mm sound motion picture in color which dramatizes the functions and benefits of air conditioning. Entitled Service Unseen, the 22 mm film covers the history and principles of air conditioning and animation is used to demonstrate

because contactor parts are ACCESSIBLE

Removing a single key permits the entire moving contact assembly to be withdrawn for inspection of all main silver contacts, both moving and stationary.

And it's easier to replace main stationary contacts, too, due to the use of individual spring retaining clips.

Write for Bulletin 4110. It fully describes Ward Leonard's new Size 1, 2 and 3 across-the-line, nonreversing AC magnetic starters. Ward Leonard Electric Co. 24 South Street, Mount Vernon, N. Y. Offices in principal cities of U. S. and Canada.

WARD LEONARD ELECTRIC COMPANY
Result-Engineered Controls
RESISTORS • RHEOSTATS • RELAYS • CONTROL DEVICES

SERVANT OF MECHANICAL AGE INDUSTRY

THE AIR YOU BREATHE SHOULD BE AS PURE AS THE WATER YOU DRINK

6 WAYS

to reduce labor and material costs

ILG VENTILATION

REG. U. S. PAT. OFF.

1. No time-wasting alignment necessary—with ILG Direct-Drive Centrifugal Fans, just slip motor into supporting motor ring and bolt it into place.

2. No special motor bases to require extra work.

3. No belts or pulleys to increase labor and material costs—install ILG fans on floor, wall, or ceiling.

4. No belt or chain guard to buy and install—fan wheel mounted directly on to motor shaft.

5. No chance of motor overload when resistance is less than calculated with "BC" type.

6. No need for excess horsepower. Sound isolation and air control are quick, easy, economical.

NEW CATALOG—illustrates and describes all lines of Centrifugal Fans. Send coupon or phone.

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the four basic functions. In addition, a number of industrial applications are shown. Prints of the film are available at no charge for showings by organized groups. . . . Three well known gas appliance manufacturing firms, each a subsidiary of Dresser Industries, have been combined into a single unit, **Affiliated Gas Equipment Co. Inc.,** Cleveland. Now under single management are **Bryant Heater Co.,** Cleveland, **Day & Night** Mfg. Co., Monrovia, Calif., and **Payne Furnace Co.,** Beverly Hills, Calif. The new firm will market a complete line of gas equipment for space heating, water heating, and air conditioning. Lyle C. Harvey, general manager of the Bryant Heater division, is also general manager and a director of the new firm.

The home office of **Pittsburgh Corning Corp.** has been transferred from the Pittsburgh Plate Glass Bldg. to new and larger quarters at 307 Fourth Ave., Pittsburgh. . . . A series of five regional air conditioning application schools for key representatives from 45 districts across the nation has been initiated by **Frigidaire Div., General Motors Corp.,** Dayton, Ohio. Each school is of one week duration and covers packaged and central conditioners. . . . **Fasco Industries Inc.,** Rochester, N. Y., the new name for **F. A. Smith Mfg. Co., Inc.,** conforms to the company trademark. There is no change in corporate structure, officers, directors, or policies.

Schaible Co., Cincinnati, has acquired the business, plant, and all facilities of **D. T. Williams Valve Co.,** also of Cincinnati. Expanded operations of the Williams firm will be conducted in the industrial high pressure valve market and the corporate status of the new division will be maintained. . . . The stud welding equipment of **Nelson Stud Welding Div., Morton Gregory Corp.,** Lorain, Ohio, can now be obtained on a rental basis in principal cities through company field engineers. . . . An addition to the building in Pittsburgh owned by **Foxboro Co.,** Foxboro, Mass., and occupied by its branch office is now under way. The new addition will be used for the assembling of control valves and for warehousing company products.

A new metal roll for the production of pipe, well casing, and general metal rolling is being manufactured by **Valley Foundry & Machine Works, Inc.,** Fresno, Calif. A heavy duty machine of the three roll, initial type, it is said to be capable of rolling heavy gage and alloy steels. The frame of the machine is all steel, and rolls are forged 4140 alloy steel. Gears are heat treated and hardened, and a specially wound, three phase gearhead motor or a variable speed slip ring motor with separate gear box is used. Sizes and lengths of these units are available to suit particular requirements.

"Eutecrod" 1700 is said to be an unusually "thin flowing" high silver brazing alloy designed for use with stainless steel, copper, brass, bronze, tungsten carbide, and steel. According to the manufacturer, **Eutectic Welding Alloys Corp.,** 40 Worth St., New York 13, it gives an excellent color match to stainless steel and is particularly recommended for making lap, butt, square, and tee joints and for sealing and filling small holes and cracks. It bonds at 1020 to 1250 F, has a tensile strength of 64,000 to 68,000 psi, and is available in five sizes from 1/64 to 3/32 in. diameter.

WHO'S WHAT

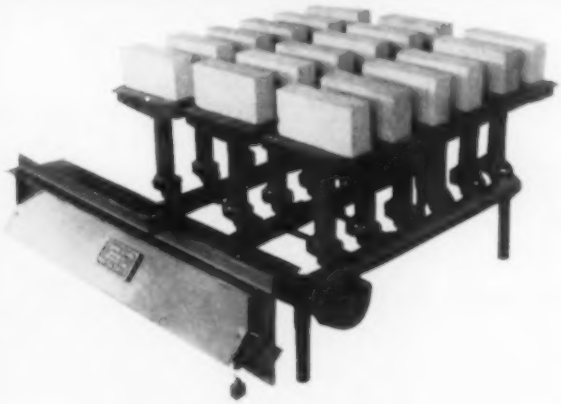
Frank H. Stohr, former executive vice president and a director of Elliott Co., Jeannette, Pa., has succeeded R. W. Davis as general manager of the Norwood works of **Allis-Chalmers Mfg. Co.,** Milwaukee. Mr. Davis, who has been with the company for 40 years, continues in an advisory capacity. . . . **James A. Greer** has resigned as advertising and sales promotion manager of **Taylor Forge & Pipe Works,** Chicago. **R. W. Murphey** is the new advertising manager.

Yarnall-Waring Co., Philadelphia, has made a number of personnel changes in its sales and manufacturing divisions. **Joseph Kildare,** sales manager, has been elected vice president in charge of sales, **Frank W. Miller,** works manager, is now vice president in charge of manufacturing. **Carl Liberg,** formerly in the company's New York sales office, is located at the new district office in Buffalo, and **C. V. Peterson** has been appointed sales representative in the Cleveland district. Numerous other changes have been made in the firm's list of steam trap distributors.

E. A. Cline, The Trane Co., La Crosse, Wis., is the new president of the **Convector Manufacturers Association. Louis D. Mandell,** C. A. Dunham Co., Chicago, has been named vice president, and **J. H. Smart,** Tuttle & Bailey, Inc., New Britain, Conn., is secretary. . . . Formerly sales manager of Deepfreeze Corp., **Frank A. Mitchell** has been named sales manager of the unit air conditioner division of **Fedders-Quigan Corp.,** Buffalo. He has succeeded **E. A. Bonneville,** recently resigned. **William G. Loye** has been appointed a sales representative of the company's heating and refrigeration divisions. With headquarters in South Bend, Ind., he will cover the northern counties in that state.

A series of changes and promotions in the sales division of **Fairbanks, Morse & Co.,** Chicago, have been effected. **O. O. Lewis,** until recently assistant sales manager, has been promoted to sales manager. **Harry L. Hilleary,** for the past 17 years manager of the firm's St. Louis, Mo., branch, is now assistant sales manager with headquarters in Chicago. **L. A. Weom,** manager of the pump division, has succeeded Mr. Hilleary in St. Louis, and **Donald T. Johnstone,** assistant manager of the pump division, has taken over Mr. Weom's former post. . . . **Allegheny Ludlum Steel Corp.,** Pittsburgh, has appointed **C. A. Scharschu** assistant technical director. His former position, director of research, has been taken over by **Dr. L. C. Hicks,** former associate director.

Ardun Co., 317 S. Detroit Ave., Tulsa, Okla., is now a distributor for **Tube Turns, Inc.,** Louisville. The firm is a partnership formed by two Tulsans, **H. W. Arduser** and **Robert H. Duenner, Jr.** . . . Vice President **Allen Butler** of **Perfex Corp.,** Milwaukee, has been named assistant to the president. **V. P. Black,** formerly manager of advertising, sales promotion, and sales training, Airtemp Div., Chrysler Corp., has been made a vice president to fill the vacancy left by Mr. Butler, as manager of the controls division.



JOHN ZINK
– Series VR –
GAS BURNER

This burner operates with high efficiency at either low or high capacities on low gas pressures.

Heat release is increased 20% to 40%. At four ounces natural gas pressure it gives 325,000 B.T.U./sq. ft. of burner area. Multiple burner heads can be arranged to fit any firebox; provision is made for standard size firebrick baffles. Burner is furnished with a blowout proof pilot, either with or without flame failure feature.

VR Burners are especially designed for boilers having low draft and small combustion space.

Ideal For Complete Automatic Control Application

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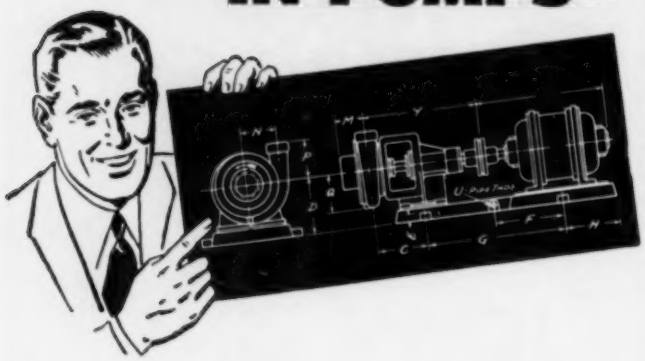
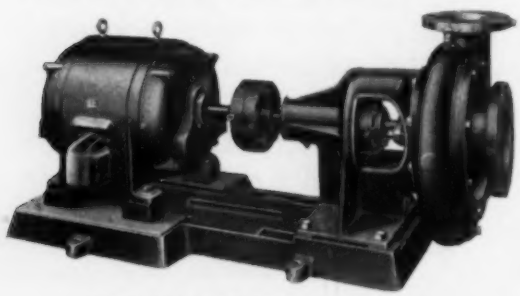
"Career Men IN PUMPS"



● *Every* key man in the Weinman organization from General Manager to Foremen in all Departments,— Design,— Engineering,— Pattern,— Foundry,— Tool Development,— Machine Shop,— Assembly Line,— Testing,— Shipping,— Sales,— has had at least twenty-five years of concentrated experience with *one* product — PUMPS. Whatever your fluid handling problems may be, it will pay you to talk with Weinman. This consultation is free for the asking . . . write, wire or phone.

WEINMAN GENERAL SERVICE UNIPUMP
May be mounted in any position . . . for use where space is limited. Sizes to 1000 G.P.M. . . . heads to 500 ft. . . . write for Bulletin 726-C1.



WEINMAN TYPES GB AND KB CENTRIFUGAL PUMPS
Side suction . . . discharge opening may be put in any one of 4 positions. For capacity and head selection, write for Bulletin 727-B.

Built by Centrifugal Specialists
The WEINMAN Pump Mfg. Co.
284 Spruce Street • Columbus 8 • Ohio

Formerly eastern regional sales manager for **Brown Instruments Div., Minneapolis-Honeywell Regulator Co.**, Philadelphia, **O. B. Wilson** is now serving as manager of sales for the east, southeast, and central regions. Related transfers of industrial managers in charge of company branch offices include: **Warren W. Matlack** from Albany, N. Y., to New York City, succeeding Mr. Wilson; **Devon W. Fryback** from Hartford, Conn., to Albany; **D. Lyle Russell** from Boston to Hartford. The firm's New Jersey sales headquarters is now a district office at East Orange and is in charge of **Herman Hoyler** who has been with the company for 11 years.

The **National Credit Men's Association** has appointed **C. G. Klock,** manager of credits and collections of the air conditioning department, General Electric Co., Bloomfield, N. J., as program chairman of the plumbing, heating, refrigeration, and air conditioning section for the NCMA national convention next May. Also named to the program committee are: **John G. de Pass,** National Radiator Co., Johnstown, Pa., vice chairman; **E. H. Figush,** Grabler Mfg. Co., Cleveland; **Howard C. Hartman,** L. J. Mueller Furnace Co., Milwaukee; **W. A. Polk,** Rheem Mfg. Co., New York City; and **D. F. Runyon,** Carrier Corp., Syracuse, N. Y. . . . **Sigmond Moroh,** former consultant to Research Corp., New York, is now chief engineer for **National Air Conditioning, Inc.,** Johnstown, Pa. He is responsible for all engineering phases of the company's triethylene glycol conditioner. . . . **Chapin Roberts,** manager of **Sears Roebuck & Co.'s** construction department, has retired after 37 years of continuous service.

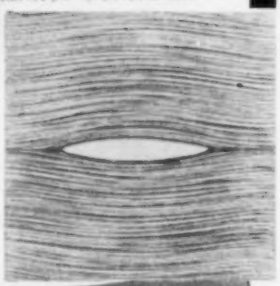
The firm of **Mel J. Stevenson and Associates,** 69 W. Washington St., Chicago, has been organized for consulting work and specializes in power plant, electrical, heating and ventilating, water and sewage system, chemical and process, and air conditioning engineering. . . . Formal opening of the new home of **Northern Heating Supplies, Inc.,** 585 New Park Ave., West Hartford, Conn., has been announced. **Harold A. Stockburger** is general manager of the company. **W. G. Lathrop** is sales manager, and **Joseph R. Howard** is sales engineer. The firm is a distributor for **York-Shipley, Inc.,** York, Pa.

Establishing a new position in the company, **Minneapolis-Honeywell Regulator Co.,** Minneapolis, has appointed **Robert H. Jacobs** superintendent of schoolhouse control sales. Handling a special line of controls, he will work directly with company salesmen throughout the country. **L. F. Lawrence** has been promoted to branch manager of the firm's Charlotte, N. C., office, succeeding **Karl Selden, Jr.,** who resigned to become sales manager for a company manufacturing textile mill equipment. **J. T. Pitts** has assumed Mr. Lawrence's former duties as senior district representative in Knoxville, Tenn. **R. T. Waits,** who has been serving on the sales staff of the Charlotte branch office, has been promoted to district representative in Greenville, S. C.

André Merle Associates, consulting engineers, are now located at 1301 Massachusetts Ave., N.W., Washington 5, D. C. . . . **J. W. LeRoy** is assistant to Carroll M. Baumgardner, vice president for sales, **National Radiator Co.,** Johnstown, Pa. He was formerly associated with Berger

PROTECT YOUR AIR CONDITIONING
INVESTMENT with
AIRFOIL OUTLETS

Airfoil
FEATURING
THE GREAT NEW *Airfoil* LOUVRE



At right is shown how the streamlined Titus Airfoil Louvre proves its superior control performance in the wind tunnel. The deep streamlined blade knifes the air like a wing of an airplane. Has little or no turbulence—absolute minimum noise—complete directional control.

THESE FEATURES MAKE AIRFOIL THE BEST BUY

- ★ "Smooth as glass," streamlined Airfoil louvre surface
- ★ Solid section louvre construction
- ★ One-piece extra strength frame
- ★ Custom-built for individual requirements
- ★ Tight sealing rubber gasket
- ★ 1¼" standard border
- ★ Positive 4-way air control
- ★ Two banks of louvres
- ★ Individual blade adjustment

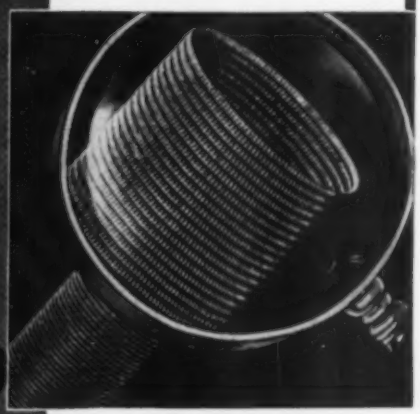
Let there be no doubt about it—Titus Airfoil grilles not only protect your air conditioning investment, but actually add value by giving it a superior air diffusion performance, which was heretofore thought impossible and which cannot be duplicated by any other air conditioning outlet. That is why Airfoil grilles are the finest money can buy.

Titus Manufacturing Corporation's adaptation of the perfected Airfoil section of an airplane wing for use in the Titus Airfoil outlets is the most important advance in air conditioning outlet design today. The Airfoil grille with its special Airfoil louvre controls the air better and gives faster, more flexible diffusion. Handles air with an absolute minimum of noise and turbulence.

Write today for new, free 1949 Airfoil Air Conditioning Outlet Catalog. Learn how Airfoil Grilles can make your air conditioning system perform with a super efficiency heretofore thought impossible.

TITUS MANUFACTURING CORPORATION
WATERLOO, IOWA

PREVENT
Rust·Scale·Dirt
FROM CLOGGING STEAM EQUIPMENT



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"The Screen's the Thing"

- ***A High Grade Monel Woven Wire Basket*** with mesh fine enough to catch the dirt—yet ample free area for passage of clean condensate, oil and other fluids.
- ***Body Cadmium Plated for Protection Against Corrosion and for better appearance.***
- ***Readily Removed Steel Blow-Off Bushing.***
- ***Bushing Automatically Aligns Screen.***
- ***Thousands in Use—Sold by nearly 150 Mill Supply Houses.***
- ***6 Sizes ½" to 2" for Pressures up to 600 Lbs.***

See Your Supply House or write for Bulletin S-201
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YARWAY STRAINERS

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Moisture Destroys Insulation Values

"Moisture increases the rate of heat transfer through a material, because water, which fills the pores or voids, conducts heat more rapidly than air." . . . *Third edition of "Insulation," by Paul D. Close as Technical Secretary of Insulation Board Institute and former Technical Secretary of the American Society of Heating and Ventilating Engineers.*

Infra Insulation is impermeable to water vapor

That is one reason why Infra—the multiple sheet aluminum insulation—is the best way to insulate even a basementless house. It is non-condensation-forming and easy to install. In crawl spaces; for example. Infra prevents heat from entering or escaping a building through ceiling, wall or floor. Observe, for instance, the remarkable results for DOWNWARD heat flow in the table of thermal factors below. These values are permanent. Infra will remain in place and not tear at the staples or elsewhere because of moistened paper and excessive weight.

Thermal Factors Printed on Every Infra Carton

Infra C Factors and Rockwool Equivalents

C.052 Heat Flow Down, equals 6" Rockwool.
C.083 Heat Flow Up, equals 4" Rockwool.
C.10 Lateral Heat, equals 3-1/3" Rockwool.

The scientific construction of Infra, with multiple separated aluminum sheets, provides 4 reflective spaces and 4 reflective surfaces, each non-condensation-forming and 97% effective against heat rays. The two sheets of aluminum and the accordion partition block convection. The two rows of inner, alternating triangular air spaces, and the small mass eliminate conduction as a problem.

WRITE FOR FREE 32-Page Booklet:
"Simplified Physics of Thermal Insulation"

Architects and engineers use it as a handbook, and colleges as a text, on heat transfer, condensation, vapor, mold, radiant heating, etc. Contains master chart of *k*, C, R and U factors of all insulations, of all thicknesses, weights, densities, etc. Address Dept. AC.

TRIANGULAR REFLECTIVE AIR CELLS AND MULTIPLE ACCORDION ALUMINUM

Infra INSULATION, INC.
10 Murray St., N. Y., N. Y.

Mfg. Div., Republic Steel Corp. . . . Representation in Tennessee, Alabama, and Georgia for the air diffusion panels of **Pyle-National Co.**, Chicago, is being handled by **Heating Equipment Co.**, 12 Third St., N.E., Atlanta. . . . **James S. Weaver** has become associated with **Culbert Pipe & Fittings Co.**, Jersey City, N. J., as a special sales representative in the New York territory.

Clyde B. Lynde is now serving as vice president and general manager of the insulation division, **Eagle-Picher Co.**, Cincinnati. **Dr. A. Paul Thompson**, as director of research, fills the vacancy created by the death of **Earle W. McMullen.** . . . At the recent annual meeting of the **Midwest Stoker Association**, **E. W. Jones**, office manager of the Chicago factory branch, Iron Fireman Mfg. Co., was elected president. Other officers elected include: vice president—**F. H. Herndon**, president, Herndon Sales & Service Co.; and secretary-treasurer—**P. I. Bohmann**, district manager, U. S. Machine Corp.

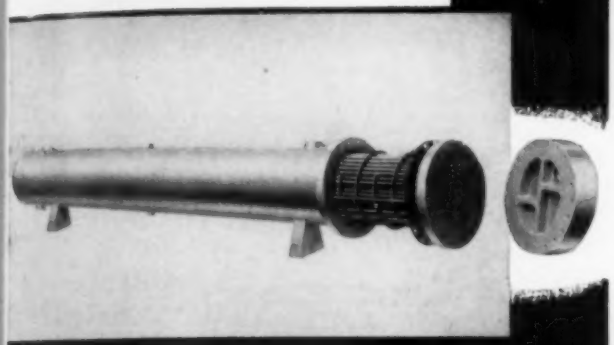
SKF Industries, Inc., Philadelphia, has appointed **Stuart H. Smith** Cincinnati district manager. He joined the company in 1934 and has been assistant district manager of the Detroit office since 1944. Newly named field representatives include: **C. N. Benson** and **D. B. Eden**, Boston; **A. R. Ehrnschwender**, Cincinnati; **J. T. Paradise**, Atlanta; and **G. L. Hansen**, Portland, Ore. . . . After three years as a partner in the firm of Bodine & Carter, **John H. Carter** is now with **Sverdrup & Parcel, Inc.**, St. Louis, Mo., consulting engineers. . . . **Edward Kellie**, president, Aminco Refrigeration Products Co., is the new chairman of the flow control valve section of the **Refrigeration Equipment Manufacturers Association.**

To help in mobilizing productive effort in the event of another emergency, the **National Resources Security Board** has enlisted the cooperation of many informal task groups from specialized segments of industry. In the group representing manufacturers of temperature and combustion controls and heating system specialties are: **L. W. McCrea**, Hoffman Specialty Co.; **C. W. Nessell** and **W. L. Huff**, Minneapolis-Honeywell Regulator Co.; **L. H. Hobson**, sales manager, automatic heating division, General Electric Co.; **Harold Marshall**, Warren Webster & Co.; **A. E. Hess**, managing director, Oil-Heat Institute of America; **Walter R. Crowe**, construction materials division, NSRB; **H. F. Collins**, Sarco Co., Inc.; **J. A. Rodgers**, president, White-Rodgers Electric Co.; **Malcolm E. Henning**, Penn Electric Switch Co.; **V. R. Tate**, executive vice president, Perfex Corp.; and **J. P. Kottcamp**, NRSB's director of the construction materials division.

With headquarters at the company's Chattanooga, Tenn., plant, **E. J. McFadden** is general sales manager of the water heater, range boiler, and soil pipe department, **Combustion Engineering-Superheater, Inc.**, New York City. . . . The **Institute of Gas Heating Industries, Inc.**, Los Angeles, has elected new officers which include: **Walter R. Dicus**, French Heating Co.—president; **T. R. Bridges**, Williams Radiator Co.—vice president; **Harry F. Haldeman**, Harry F. Haldeman, Inc.—secretary-treasurer; **A. J. Horn**, Payne Furnace Co., **Arthur T. Enderle**, Frank X. Enderle Co., and **Frank T. Daneil**, Home Heating Service Co.—directors. **Ernest W. Kimmell** continues as managing director.

PATTERSON-KELLEY
Heat Exchangers
AND
Process Equipment

Heat-Transfer
SPECIALISTS



Specializing
IN REFRIGERANT
COOLERS

The very essence of "air conditioning" is the controlled transfer of heat.

It was entirely logical, therefore, that Patterson-Kelley engineers, who have specialized for years in heat-transfer equipment, should turn their attention years ago to refrigerant coolers, condensers and other heat-transfer equipment needed in air conditioning and refrigeration systems.

Today, Patterson-Kelley Equipment, coolers for example, is considered by many to be the standard for such installations.

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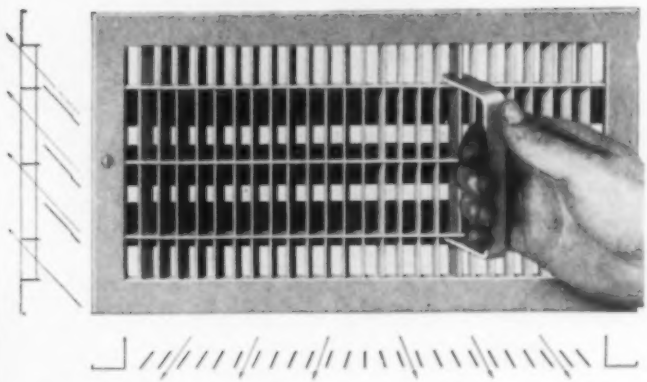
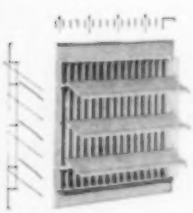
INDEPENDENT
"Fabrikated"
WALL GRILLES WITH
DEFLECTING VANES



Style 321-A grille with deflecting vanes



Rear view showing adjustable deflecting vanes

ACCURATE and positive compound deflection of air flows can be readily secured with these "Fabrikated" grilles. Directional adjustments may be made when grilles are installed; or grille bars and vanes may be adjusted after installation to meet unforeseen or changed requirements.

Each grille bar and each deflecting vane is adjusted *individually* with a special two-pronged tool included with each shipment. Locking of grille bars and vanes is not required because they are held firmly in place — no vibration — no rattle.

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DY-NAMIC BALANCING

★ BOOSTS MAINTENANCE EFFICIENCY

★ PROLONGS BEARING LIFE

★ SAVES VALUABLE EQUIPMENT

★ CUTS MAINTENANCE COSTS

THE QUAKER OATS CO., Maintenance Shop, Cedar Rapids, Iowa, reports: Our "Bear" Dy-Namic Balancing Machine Model 375-R will do everything we ask in the way of attaining Dy-Namic Balance for hammer mill rotors, fan wheels, pulleys, armatures, etc. Daily it uncovers destructive Dy-Namic Unbalance up to 2 lbs., thus enabling us to prevent excessive vibration in valuable machinery and premature wear of bearings and equipment.

"Bear" Dy-Namic Balancing Machines are "naturals" for maintenance applications. Adjustable columns enable the operator to handle various length parts readily. The adjustable rollers provide ease of adjustment in changing the machine to suit various shaft diameter requirements. The universality of the "Bear" Machines permits a wide variety of applications with a minimum of time required for set-up changes.

"Bear" Machines are now being successfully used in numerous Heating, Piping and Air Conditioning applications for maintenance work, as well as in the manufacturing process. Some typical applications are:

Air Conditioning Fans — Ventilating Fans
Blower Wheels for Oil Burners — Draft Fans
Centrifugal Pumps — Refrigeration Units

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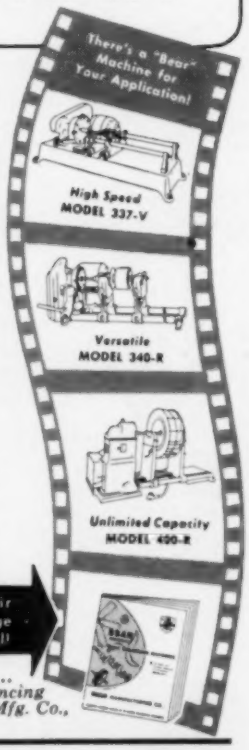

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STATIC AND DY-NAMIC BALANCING MACHINES

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2965

Vapor Heating Corp., Chicago, has recently opened its 13th sales office. This one, in Houston, Tex., is in charge of **J. T. Elwood,** formerly of the firm's St. Louis branch. . . . A branch office at 123 W. College Ave., Appleton, Wis., has been opened by **Foxboro Co.,** Foxboro, Mass., and **Milton A. Schreiner,** with the company since 1922, is the manager. **George J. Willier** is in charge of another new branch office at 6225 Brookside Ave., Kansas City, Mo. At Baton Rouge, La., **John B. Deaderick** is managing the office recently opened at 115 N. 19th St. In addition to these fully staffed branches, resident engineers have been quartered in two cities; **Eugene F. Machell** in Milwaukee, and **Russell E. Steward** at Frankfort, N. Y., serving the Utica area.

Directors of the **Canton Pattern & Mfg. Co.** and **Canton Stoker Corp.,** Canton, Ohio, jointly operated concerns, have elected **LeRoy C. Dubs** president, to succeed his late father, Theodore M. Dubs, founder and president of both companies. **Glenn T. Dubs,** youngest son of the founder, is treasurer of the two firms and is general manager of the pattern and manufacturing company. Other newly elected officers include **William I. Zink,** vice president of both concerns and **Fred Eisenberger,** a director of both and production manager of the pattern firm. Appointments in the stoker company include those of **Donald J. Sprague,** chief engineer and production manager, **Harold Buchtel** as office sales manager, and **Walter Sherman** as factory superintendent.

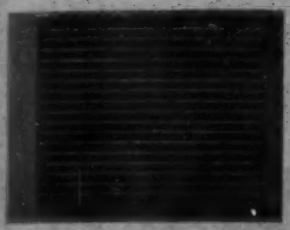
Stewart Kerr, a Detroit attorney who has served as special assistant to the United States Attorney General, is now executive secretary of the **National Association of Engineering Companies.** . . . New directors of **Reinhard Brothers Co.,** Minneapolis, include **Victor J. Hillstrom,** manager of the North Dakota division since 1937, **Henry C. Hoffman,** manager of the South Dakota division since 1947, **R. M. Wagner,** with the firm two years and new treasurer, **James L. Pause,** new vice president in charge of home appliance sales, **George C. Scott,** new vice president in charge of automotive sales, and **Harry E. Latourell,** new vice president in charge of plant operations. **E. L. Davis,** former vice president and treasurer, has succeeded President **A. C. Reinhard** who has become chairman of the board of directors.

The 29 year old **Sheffler-Gross Co.,** Philadelphia, is the exclusive sales representative for **McAlear Mfg. Co.,** in that area. . . . Recently congratulated by **Frigidaire Div., General Motors Corp.,** Dayton, for topping the 1948 records of 45 sales districts were **E. A. Blackburn,** of **Cox and Blackburn, Inc.,** Houston, Tex. distributor, and **D. R. Steele,** Houston commercial sales manager. . . . The **Refrigeration Contractors Association of Los Angeles** has elected **Ralph E. Manns,** of Wilmington, president. Other new officers include **Hal Crumly,** Pomona, vice president, and **Ralph M. Westcott,** Los Angeles, secretary-treasurer. **Neal S. Templin** remains executive secretary.

Peerless Pump Div., Food Machinery and Chemical Corp., Los Angeles, has formed a new sales district with headquarters at 4330 Leavenworth St., Omaha, Neb. District manager is **J. W. L. Stone** whose office will cover all or part of 11 western and midwestern states. . . . **Frank Haas,** president, Haskris Co., is now president of

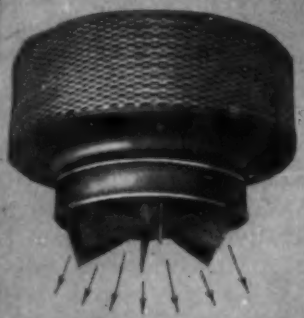
McCORD
HEATING AND AIR CONDITIONING PRODUCTS

BLAST HEATING COILS—AIR CONDITIONING COILS—ALL TYPES



All copper spiral fin tube construction with the tubes formed or bent permits uniform expansion without strain. Headers are heavy wall construction with tubes brazed into extended ferrules formed in the headers. Fins are helically wound and metallically bonded. May we send catalog?

VERTICAL TYPE UNIT HEATERS



A proven heating unit, modern in design, incorporating every unit heater advancement. McCord copper spiral fin tube construction creates air turbulence without undue restriction. Improved fan blades deliver large volume of air with minimum air noise. Guaranteed for 150 lbs. saturated steam pressure. Specify McCord.

HORIZONTAL TYPE UNIT HEATERS

A time proven heating unit, modern streamlined in design, quiet, yet its rugged construction adapts it to all types of installations, guaranteed for use on No. 150 p. s. i. steam pressure. All copper heating element, individual spiral fin tubes free to expand without strain on joints. Standard motors with standard bases used. Wide range of capacities.

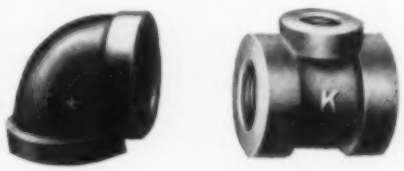
Save fuel with McCord Unit Heaters.

McCord CORPORATION
DETROIT 11, MICH.

Time was saved "at every turn" because "K" Fittings made up fast. This upped the profit considerably above the estimate.

Because "K" Fittings are chamfered and tapped on precision machines, pipes enter and screw in easily. In the case of "K" flanged fittings, the faces are accurately milled and drilled. Fittings are carefully inspected and gaged.

Ask your supplier for "K" Fittings.



The "K" line of 3000 shapes and sizes of cast-iron fittings includes:

- Standard and Extra Heavy Screwed Fittings
- Standard and Extra Heavy Companion Flanges
- Standard Flanged Fittings
- Drainage and Sprinkler Fittings

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KUHNS BROTHERS CO.
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COMBINED MARKET FACILITIES with Malleable Iron Fittings Company, Branford, Conn., and Kuhns Bros. Co., Dayton, Ohio

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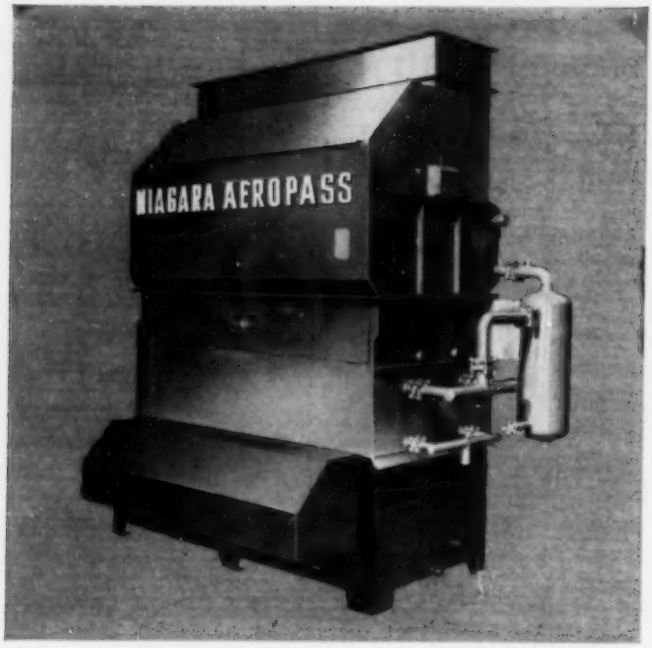

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New Condenser Cuts Refrigeration Costs Saves Cooling Water

● The Niagara Aeropass Condenser cuts the cost of refrigeration by running compressors at lower head pressure, saving up to 35% of power.

The refrigerant gas passes thru two coils in an air stream. The first, "Duo-Pass"* dry coil, removes the superheat by air cooling and condenses oil vapor. The second, condensing coil, drenched by recirculated water spray, condenses by evaporation, transferring to the air 1,000 BTU for every pound of water evaporated and saving more than 95% of the water used by water-cooled condensers. This done at low temperature, no scale forms on condenser tubes to clog air passage.

Between the two coils is the "Oilout"*, which purges the system of crankcase oil and dirt, keeps it always at full capacity.

The "Balanced Wet Bulb"* control holds head pressure low, automatically giving the full benefits of power saving in cool weather and providing always full capacity for peak loads.

Niagara Aeropass design results from over fifteen years' experience condensing by air. It is completely trustworthy for year' round operation. Users say, "It saves half the difficulties and labor of running a refrigeration plant."

Units range from 10 to 100 tons capacity.
For full information ask for Bulletin 103.

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INDUSTRIAL COOLING HEATING • DRYING
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HUMIDIFYING • AIR ENGINEERING EQUIPMENT

the **Refrigeration Contractors Association of** Chicago, **Walter McCarty**, president McCarty Bros. Equipment Corp., is the new vice president, and the secretary-treasurer is **Alphonse Gerat**, of A. G. Refrigeration Sales & Service, Inc.

Daniel Mapes, vice president in charge of engineering and development and a director of **Walter Kidde & Co., Inc.**, New York City, is serving as a vice president of the **Compressed Gas Association.** . . . **R. S. Reynolds, Jr.**, of Reynolds Metals Co., Richmond, Va., is president of the **Aluminum Association.** Recently elected vice presidents of the association include: **E. G. Grundstrom**, Advance Aluminum Castings Corp., Chicago; **M. E. Rosenthal**, United Smelting and Aluminum Co., Inc., New Haven, Conn.; and **George N. Wright**, John Harsch Bronze and Foundry Co., Cleveland. **A. V. Davis**, Aluminum Company of America, New York, has been re-elected chairman of the board and **Donald M. White** was re-appointed secretary-treasurer.

After three years as promotion manager of the Charlotte News, **George V. Hunter** has joined **Radiator Specialty Co.**, Charlotte, N. C., as advertising manager. . . . **Bror G. Dahlberg**, chairman of the board of directors, Celotex Corp., Chicago, is the new chairman of the board of governors of the **Building Officials Foundation**. Other new board members are **Harold Boeschenstein**, president, Owens-Corning Fiberglas Corp., Toledo, and **Henry Bohnsack**, president, International Steel Co. Evansville, Ind. . . . **R. J. Schuler** is vice president of **Cauhorn Distributing Co.**, Detroit, handling Kaiser aluminum sheets among other products. **A. V. Cauhorn Co.** was recently named a factory distributor for **Airtemp Div. of Chrysler Corp.**, Dayton. Mr. Schuler continues to supervise sales activities of both Cauhorn companies.

During the past four years, **W. H. Magee** has been midwest sales manager for U. S. Gauge Co., Division of American Machine and Metals, Inc. He is now general sales manager for **Russell Electric Co.**, Chicago, subsidiary of Raytheon Mfg. Co. . . . The **Refrigeration Industry Safety Advisory Committee** has named **Cyrus W. Miller** as full time secretary and safety code consultant, with offices at 155 E. 44th St., New York City. The committee is sponsored jointly by representative groups of manufacturers, including the Air Conditioning and Refrigerating Machinery Association, the Refrigeration Equipment Manufacturers Association, the National Electrical Manufacturers Association, and the Compressed Gas Manufacturers Association.

Frank D. Peltier is now director of engineering, appliance division of the engineering department, **Philco Corp.**, Philadelphia. **Matson C. Terry** is chief engineer in charge of developing single room air conditioners. . . . As manager, heating products, **Jones & Brown, Inc.**, Pittsburgh, **David M. Ramsey** is responsible for the sales and distribution of the firm's conditioning units and controls. He was formerly district sales manager, southeastern territory, of Richmond Radiator Co. . . . **Frank E. Early**, with **Askania Regulator Co.**, Chicago, since 1936, is representing the firm in the Detroit area and is now located at 15234 Michigan Ave.

In a Word . . . DEPENDABLE!

HERE is one of several Zallea Stainless Steel Expansion Joints installed in the 250 lb. SWP 450° F. main steam distribution line of the Alan Wood Steel Co., Swedeland, Pa. . . . put there to assure uninterrupted production—and a continuous flow of steam to the mill. This installation is *another* vote of confidence for Zallea Stainless Steel Expansion Joints as the one ideal, *dependable* medium for absorbing movement in pipe lines due to thermal changes. Check these features to see why more and more engineers and designers insist on Zallea when they want the best:

- *More economical than expansion bends*—the cost of insulating and installing a bend alone is frequently more than the cost of a Zallea Stainless Steel Expansion Joint.
- *No maintenance*—having no packing they may be used in underground lines without manholes.
- *Low thrust against anchors*—require only 1/10 the force required to compress some slip type expansion joints.
- *An expansion joint for every type of service*—special types for combined axial and lateral movement.
- *Shorter face to face dimensions*—enables installation in close quarters.
- Hydraulic method of forming assures uniform wall thickness throughout corrugations.
- As easy to install as an ordinary flanged or welding fitting.
- Traverse range from fractions of an inch up to 7¼" in a single unit and 15" in a double unit.
- Pressure range from vacuum to 300 lbs. in standard units. Up to 1000 lbs. in special units.
- Temperature range from sub-zero to 1600° F.

Catalog 47 contains complete information on all Zallea Expansion Joints. We would be pleased to send you a copy. Write today: Zallea Brothers, 816 Locust St., Wilmington 99, Del.

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EXPANSION JOINTS

WORLD'S LARGEST MANUFACTURERS OF EXPANSION JOINTS EXCLUSIVELY

Thermobloc

DIRECT-FIRED UNIT HEATER

Advanced

DESIGN

Automatic

CONTROL

Inexpensive

OPERATION

There is no unit heater on the market with the simplified, direct transfer design found in THERMOBLOC.

Complete automatic control offers a safety factor and saves cost of an operator.

Direct transfer of heat from the products of combustion to the air without an intermediary, utilizes every possible heat unit in the fuel, with the amazing results of 82-86% efficiency with No. 3 oil or gas.

Immediately available in two sizes:

THERMOBLOC 300

300,000 b.t.u.'s, only 7' high, weight 800 lbs.

THERMOBLOC 555

550,000 b.t.u.'s, 10' high, 30" diameter, weight 1200 lbs.

THERMOBLOC DIVISION

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Manufacturers of the well known Thermix Power Plant Equipment

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Worry-Free Service
The Most Advanced Welding Process

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Trentweld Stainless Steel Tubing

On all types of piping jobs where high temperature or corrosive action must be considered, future trouble can be avoided by installing stainless steel tubing. You know that . . . but from an engineering point of view you may not be familiar with the specific advantages of Trentweld stainless steel tubing.

To begin with, Trentweld is no sideline product. It is produced in light-weight, thin-wall form in diameters from ⅛ inch to 30 inches, by a mill exclusively devoted to stainless tubing. This work-horse tubing is rolled and machine-welded by an exclusive process that makes it tougher, structurally stronger. It is sized to make-up with standard fittings for vanstoned or welded joints, provides you with tubing that makes installation work a time-saving, smooth-running job while it delivers the service you have a right to expect.

› Call on Trentweld engineers, without obligation, for specific information, or write for the Trentweld Data Bulletin.

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NEW BOOKS & REPORTS

ASTM Pipe Specs Include New Tentative Standards

First issued in its present form in 1942, the booklet Specifications for Steel Piping Materials, published by the American Society for Testing Materials, needs no introduction. Last year an editorial committee reviewed all of the pipe and tubing specifications and as a result, improvements and constructive editorial changes have been incorporated in the latest (sixth) edition.

Chief differences from the previous edition will be found in the newer tentative standards dealing with seamless carbon-molybdenum and chromium-molybdenum alloy pipe for high temperature service, seamless and welded austenitic stainless pipe, and both carbon steel and alloy steel bolting materials.

Among the pipe specifications are those dealing with welded and seamless pipe, electric-fusion-welded steel pipe, welded alloyed open hearth iron pipe, seamless chromium-molybdenum alloy steel pipe for service at high temperatures, and seamless and welded austenitic stainless steel pipe.

Under the heading boiler, superheater, and miscellaneous tubes, there will be found specifications for lap welded and seamless boiler tubes, welded alloy steel boiler and superheater tubes, copper brazed steel tubing, seamless and welded austenitic stainless steel tubing for general service, and others.

There are three specifications covering refinery still tubes, and three covering heat exchanger and condenser tubes. There are five specifications on castings, mild- to medium-strength carbon steel for general applications, carbon steel for valves, flanges, etc. (for high temperature service), alloy steel for valves, flanges, etc. (for high temperature service), carbon steel suitable for fusion welding for high temperature service, and alloy steel for fusion welding for high temperature service. There are also specifications for forgings and welding fittings; four cover bolting; and one on grain size.

Copies of this 330 page book, heavy paper cover, can be obtained for $3.00 from ASTM headquarters, 1916 Race St., Philadelphia 3.

Standard Covers Mineral Wool for Low Temperature Use

Commercial Standard CS105-48, Mineral Wool Insulation for Low Temperatures (second edition), covers minimum physical and chemical requirements of mineral wool in loose, granulated, felt, industrial batt, board and block forms. It includes thicknesses of insulation required for various operating temperatures, specifications for auxiliary materials, tests, installation requirements and method of guaranteeing compliance with the standard.

The booklet, prepared by the National Bureau of Standards, also includes a brief history of this standardization project, a list of the acceptors, and the membership of the standing committee, the chief function of which will be to consider subsequent revisions.

Printed copies may be obtained from the Superintendent of Documents, Government Printing Office, Washington 25, D. C., for ten cents. A discount of 25 percent will be allowed on orders for 100 or more copies sent to a single address.

As advertised in TIME and NEWSWEEK

AIR-MAZING FACTS

1. BREATHING HEAVILY! The air you breathe in a week's time weighs more than you do!

2. WHAT'S YOUR TYPE? Hundreds of types of filters to remove dirt from the air and contamination from liquids have been developed by Air-Maze, the filter engineers.

3. AIR-MINDED HOTELS! To give you the utmost in comfort, thousands of leading hotels, restaurants, trains and office buildings keep their air clean and pure with Air-Maze filters—first choice of ventilating engineers and architects.

WHETHER YOU build or use engines, compressors, air conditioning and ventilating equipment, *or any device using air or liquids* —the chances are there is an Air-Maze *engineered* filter to serve you better. Representatives in all principal cities, or write Air-Maze Corporation, Cleveland 5, Ohio.

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The Filter Engineers

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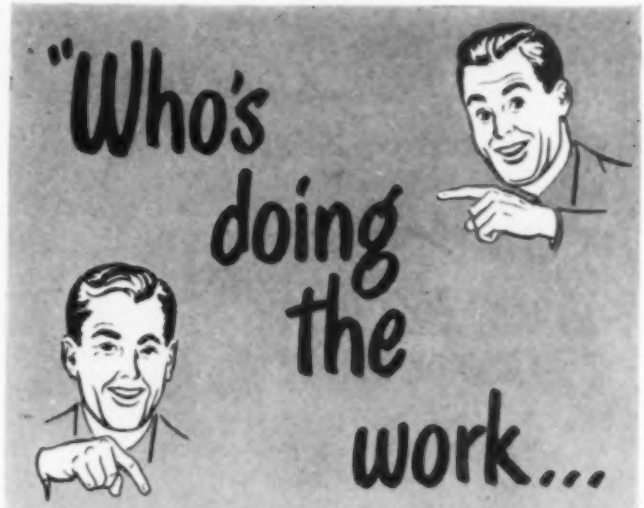
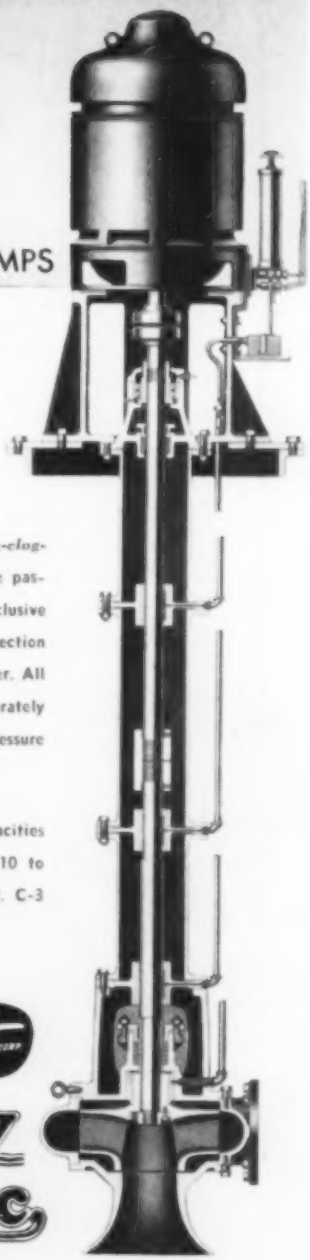
SILENCERS
SPARK ARRESTERS
LIQUID FILTERS
OIL SEPARATORS
GREASE FILTERS

"Who's doing the work... YOU OR THE PUMP?"



SAVE TIME... SAVE MONEY... ON MAINTENANCE WITH ECONOMY PUMPS



Maintenance time and costs hit a new low when you do your pumping with Economy. Case histories on Economy's Sump Pumps, for example, show continuous trouble-free operation over thousands of hours service.

Screenless, these pumps are *really non-clogging!* Smooth, clear passages permit free passage of rags, solids or bulky material. Exclusive heavy-duty shock absorber prevents deflection of shaft by heavy objects entering impeller. All joints in drop pipe suspension are accurately machined and registered. Bronze, pressure grease-lubricated bearings.

Available in single or duplex units. Capacities from 50 to 10,000 G.P.M. Heads from 10 to 100 ft. For complete details, write Dept. C-3 for Bulletin E-345.

Centrifugal, axial and mixed flow pumps for all applications.

Economy Pumps Inc.

Division of Hamilton-Thomas Corp.
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*Z-Crete is the registered trademark of the Zonolite Company.

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Classifies Fans

NAFM Standards for Classifying Air Moving Equipment is a recent four page bulletin (No. 108) which classifies fans in two general types, illustrates group classifications, and very briefly and nontechnically describes the functions of each.

The first general type of air moving equipment includes centrifugal, vaneaxial, tubeaxial, and propeller fans for ventilation and industrial use. The second general type, called electric fans, is broken down into wall or bracket fans, desk fans, air circulators or pedestal fans, and ceiling fans.

Information on specific differences in design, construction, and performance of various fans is not given. For example, the vaneaxial fan is described as suitable for moving air or gases over a wide range of volumes and pressures, whereas the tubeaxial fan is said to move air or gas through a wide range of volumes at medium pressures. However, it should be remembered that the booklet has been issued to assist the public as well as the engineer in the use of standard terminology.

Free single copies are available from NAFM headquarters, 5-157 General Motors Bldg., Detroit 2.

Safety Standards Feature Air Conditioning, Heating Systems

The pamphlet Building Code Standards of the National Board of Fire Underwriters for the Installation of Heat Producing Appliances, Heating, Ventilating, Air Conditioning, Blower, and Exhaust Systems is just what its title implies—a compilation of safety standards.

Much of the information contained was included in the board's 1943 edition of its Recommended Building Code, but is now available in a separate publication for the first time. All of the material has been reviewed and brought up to date. Copies may be obtained from NBFU headquarters, 85 John St., New York 7.

Other Books and Reports Received

Handling and Stowage of Ship's Perishable Cargoes—An eight page paper by L. L. Westling, refrigeration engineer with Matson Navigation Co. for 27 years, deals with such subjects as conditions peculiar to marine transport, classes of refrigerated ships, methods of packing and types of containers, and the temperature and humidity at which common perishables should be held. Published by the American Society of Refrigerating Engineers, 40 W. 40th St., New York 18, the paper is identified as Application Data No. 43, costs 50 cents.

The Thermal Drying of Fine Coal, by O. R. Lyons, research engineer, and A. C. Richardson, supervisor, Battelle Memorial Institute—Contains data which, the authors say, may be used with a few tests to determine the difficulty of the drying problem, the approximate cost in dollars per ton of water removed, the required exhaust gas temperature as determined by the final surface moisture content of the dried coal, the inlet gas temperature, the inlet or exhaust gas weight and volume, the type and size of dryer to use. The 23 page, paper covered report is based on research sponsored by Bituminous Coal Research, Inc., 912 Oliver Bldg., Pittsburgh 22, and reprint copies are available from BCR headquarters.

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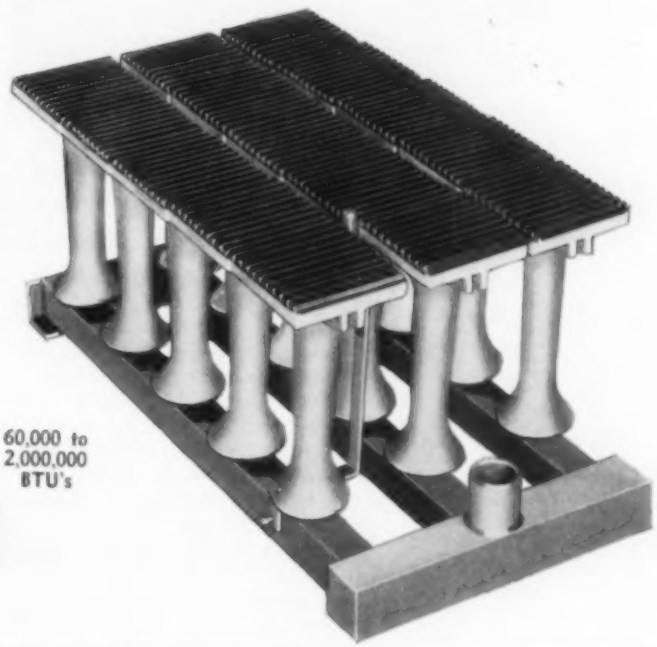

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The Reinhard 700 Series embodies all the latest developments in conversion burners for cast iron sectional and steel fire box boilers. The cast burner heads employ the Up-Shot principle and are made to withstand tremendous heat. Quiet in operation, they make maximum use of fuel to give real economy of operation.

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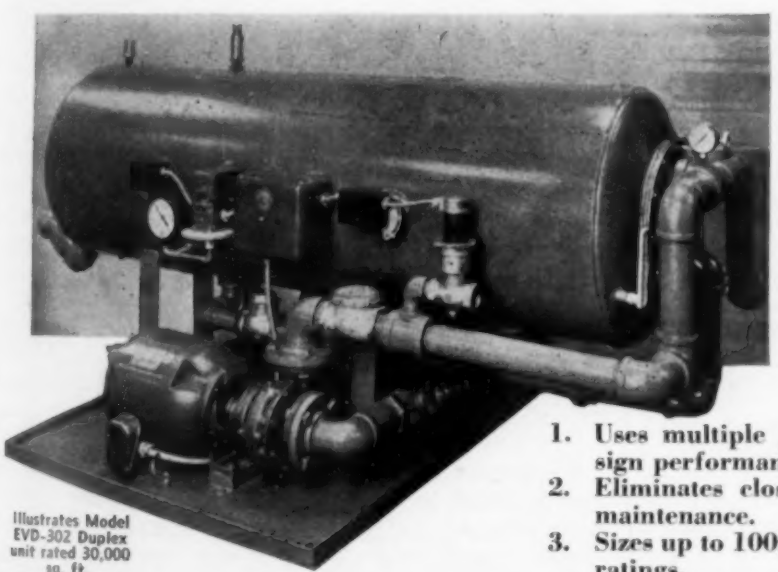

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1. Uses multiple jet principle, prolonging indefinitely design performance.
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MEETINGS & CONVENTIONS

Heating and Air Conditioning Seminar—The first annual seminar dealing with new developments in heating and air conditioning is being held at New York University, consists of 15 weekly evening meetings (which began February 8), and is open to engineers and technical salesmen. The seminar is conducted by a panel consisting of Mario C. Giannini, associate professor, New York University, Emory N. Kemler, research professor and acting director of research, Peter B. Gordon, treasurer, Wolff and Munier, and Victor L. Parsegian, director, physical research, Kellex Corp. Topics include the heat pump, radiant panel heating and cooling, instrumentation, and controls. Additional details may be obtained from Prof. S. G. Roth of the University's division of general education, 100 Washington Square, E., New York 3.

Chicago Technical Conference and Production Show, March 14 to 17, Stevens hotel, Chicago—Some 17 technical sessions are planned and will deal with equipment, methods, shop practice, processing, management, research, plant facilities, etc. Both the show and conference are sponsored by the Chicago Technical Societies Council, 53 W. Jackson Blvd., and its 55 affiliated societies.

Heat Pump Meetings—Dr. E. N. Kemler, research professor of mechanical engineering, New York University, will discuss heat pump developments before the Morris County New Jersey Engineers Club, March 15 and the ASME student section at Rhode Island State College, Kingston, April 17.

8th Heating and Air Conditioning Conference—To be held at Iowa State College, Ames, Iowa, March 21 to 23, it is jointly sponsored by the college, the ASHVE Iowa chapter, and the Sheet Metal Contractors Association of Iowa. According to the tentative program, sessions will deal with trouble shooting, trends in institutional heating, boilers and combustion, panel heating, servicing air conditioning systems, warm air heating, and sheet metal problems.

National Warm Air Heating and Air Conditioning Association, 145 Public Square, Cleveland 14—Indoor Comfort Conferences are continuing during March and April in a number of cities located in Michigan, Indiana, Illinois, Minnesota, Nebraska, Missouri, New York, and Ohio. Each of the conferences (warm air heating schools) consists of a three day instruction program.

6th Oregon State Air Conditioning Conference—To be held March 24 to 26 at Corvallis, Ore., this affair will include five technical sessions and a banquet. It is sponsored by the Oregon chapter of the ASHVE and by the mechanical engineering department of Oregon State College.

Texas Air Conditioning Conference—The North Texas, South Texas, Southwest Texas, and Shreveport chapters of the American Society of Heating and Ventilating Engineers are sponsoring the 5th annual air conditioning conference at A. & M. College of Texas, College Station, March 28 to 30.

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Through Sheet Metal Work

with Black & Decker Porto-Shears*

Black & Decker Electric Porto-Shears cut sheet metal too tough for snips, speed up jobs where snips are slow. Fast, powerful shearing action eliminates hard muscular strain. New 12-gage unit has been developed to increase the range of the Porto-Shear line to heavier materials. Over-all design simplicity is retained with heavier construction necessary to shear 12-gage steel. Also shears about two gages thinner in Monel and stainless; 50% above rating in sheet copper, aluminum, lead, etc. Other models include 16- and 18-gage Porto-Shears.

For full information on Porto-Shears and many other speedy Electric Tools, see your nearby B&D Distributor. Write for free catalog to: The Black & Decker Mfg. Co., 628 Pennsylvania Ave., Towson 4, Maryland.

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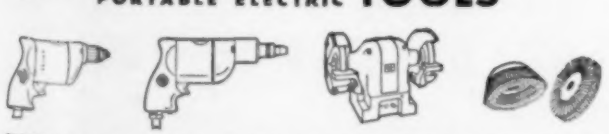

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MORE HORSEPOWER PER DOLLAR

SAVES SPACE TOO...

Though operating at 3000 r.p.m. with ratings from 1/40 to 1/10 h.p., the short length of the Raytheon type "350" Motor adapts it for use with speed reducing methods with an actual saving of space as compared to equivalent rated motors operating at 1500 to 1700 r.p.m.

The "350" can save you up to 50% in cost over comparably rated series or induction motors. Exceptionally quiet, there are no cut-out switches to fail, no capacitors to replace and no brushes to maintain.

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- Pumps
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Engineers' Council of Houston, 2615 Fannin St., Houston 4, Tex.—The second annual symposium, which will deal with the conservation of our national resources, has been scheduled for April 2, the Rice hotel, Houston. Five papers on the major phases of conservation will be presented by well known authorities.

NACE Conference and Exhibition—The fifth annual affair of the National Association of Corrosion Engineers, 905 Southern Standard Bldg., Houston, Tex., will be held at the Netherland-Plaza hotel, Cincinnati, April 11 to 14. Some 42 technical papers will be given at 11 symposia on corrosion principles, chemical, electrical, and communication industries, cathodic protection, protective coatings, salt water corrosion, and the gas, transportation, and general industries. Among the many technical papers scheduled are: Why Metals Corrode, by H. H. Uhlig, Massachusetts Institute of Technology; Engineering Aspects of Cathodic Protection as Applied to Pipe Lines, by E. P. Doremus, G. L. Doremus, and M. E. Parker, Cathodic Protection Service, Houston; Erosion-Corrosion of Metals and Alloys, by W. A. Luce and M. G. Fontana, Ohio State University; Porcelain Enamel as a Corrosion Resistant Coating, by G. H. McIntyre, Ferro Enamel Corp., Cleveland; Corrosion of Steel Pipe by Chlorinated Sea Water at Various Velocities, by V. B. Volkening, Dow Chemical Co., Freeport, Tex.; and Cathodic Protection of Gas Distribution Systems, by A. W. Peabody and C. L. Woody, Ebasco Services, Inc., New York.

The 11th Midwest Power Conference, April 18 to 20, Sherman hotel, Chicago—Sponsored by Illinois Institute of Technology, 3300 S. Federal St., Chicago 16. Conference secretary is Dr. E. R. Whitehead, director, electrical engineering dept., Illinois Tech.

New Jersey Society of Professional Engineers, 86 E. State St., Trenton 8, N. J.—25th annual convention, to be held April 29 and 30, Essex House, Newark, N. J. A major feature will be industrial, engineering, and commercial exhibits.

American Foundrymen's Society, 222 W. Adams St., Chicago—1949 annual meeting, May 2 to 5, St. Louis, Mo.

Instrument Society of America, 1117 Wolfendale St., Pittsburgh 12—The fourth annual spring meeting will be held at the Royal York hotel, Toronto, Canada, May 12 and 13. The tentative program indicates four technical sessions and a dinner. The 1949 conference and exhibit are scheduled for September 12 to 16 in St. Louis, and the 1950 convention and exhibit will be held in Buffalo, September 11 to 15.

Oil-Heat Institute of America, 6 E. 39th St., New York City—1949 national convention and National Oil-Heat Exposition, May 16 to 20. Boston. Convention headquarters will be the Statler Hotel and the exposition will be in Mechanics Hall.

Speakers will be heard on merchandising, engineering, construction, and industry problems. May 20 has been designated "engineering day".

Those planning to attend are urged to make hotel reservations at the earliest possible moment by writing

UNITED
REDWOOD "PACKAGE"
Atmospheric
COOLING TOWERS

Pre-fabricated *Easily Assembled*
Made in Standardized Sizes for
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Towers ready for shipment. Larger towers of all kinds for commercial and industrial purposes.

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FOR FAST, QUIET AIR MOVEMENT INSTALL Peerless Electric PROPELLER FANS

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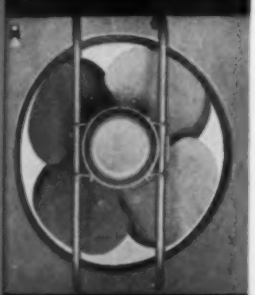

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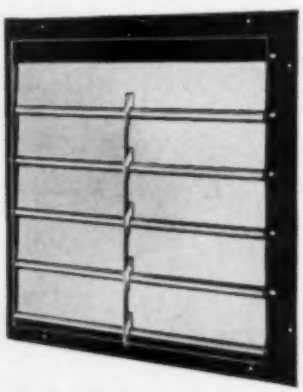
NO matter what your need for quiet, variable speed, attractively designed fans, you'll find the answer in a Peerless Electric design.

A combination of specially engineered capacitor motors and perfectly balanced extra wide blades produces extremely quiet operation. Motors are totally enclosed and permit installation in either a vertical or horizontal position.

These fans are supplied in single-speed, two-speed or three-speed operation and finished in gray. Recommended for stores, restaurants, hotels, offices and similar installations where quietness of operation and attractive appearance are prime requisites. Write for descriptive folders.



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Peerless Electric automatic louvers are designed especially for use with propeller fans. When the fan starts, the air stream instantly opens the louvers, which close by gravity when the fan stops. Completely weatherstripped around the inner edge of the frame. Built of high grade aluminum. Sizes from 10 to 48 inches.

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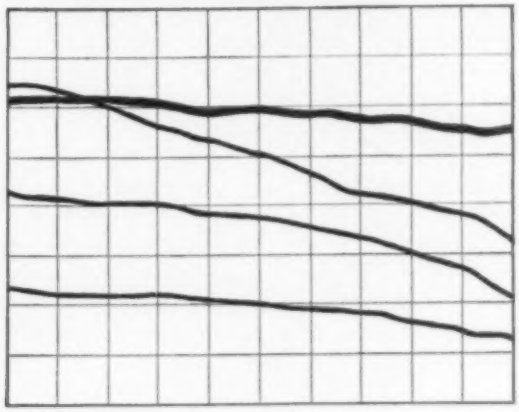
THE LOWDOWN ON INSULATION



Efficiency is as efficiency does

Remember the old saying—"Handsome is as Handsome does"? Just change that word "handsome" to "efficiency," and you're talking about low-temperature insulation.

The smart insulation buyer realizes that theoretical or laboratory efficiency isn't the whole story, by any means. He wants his insulation to keep on delivering efficiency in his cold-room walls.

One of the grand things about Novoid Corkboard insulation is the dependable way it stays efficient year after year in actual service. Our customers keep coming back for more. (Satisfaction—not force of habit—is the reason for that.)

Of course, Novoid Corkboard is made here in the United States, under strict quality control. And it's installed by some of the most experienced, dependable men in the business. It's about as good a way to keep refrigeration costs down as you can possibly find. If conflicting insulation claims are bothering you, just write to us. We'll be glad to send you the "lowdown." Novoid Cork, Englewood, N. J.

Consider EVERY factor when you buy insulation

Check list of insulation properties

	Novoid Cork Insulation	Many Other Insulations
First cost	moderate	low to high
Installation cost	low	moderate to expensive
Eventual cost	low	high
Efficiency	remains high	generally deteriorates
Moisture resistance	excellent	some good, some poor
Vapor seal	simple	complicated or difficult to install
Structural strength	very good	some non-existent, others fair
Weight	very light	varies
Resilience	considerable	from fair to complete rigidity
Settling	none	a serious problem
Supporting structure	none	elaborate
Fire resistance	good	excellent to poor
Vermin resistance	good	excellent to very bad
Odor absorption	none	some are regular sponges

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to the Boston Chamber of Commerce, housing bureau, 80 Federal St., Boston 10.

National District Heating Association, 827 N. Euclid Ave., Pittsburgh 6—40th annual meeting. New Ocean House, Swampscott, Mass., May 24 to 27.

Food Distribution Exposition—Sponsored by the United States Wholesale Grocers Association, the first national show to be devoted to all phases of the food distribution field will be held at the St. Louis Auditorium, May 30 to June 1. Wholesalers' warehouse and office operation, store modernization, and merchandising methods will feature the business sessions. Registration cards for admission may be obtained from Clapp and Poliak, Inc., 350 Fifth Ave., New York 1, show manager.

Fourth Annual Construction Industries Exposition and Home Show—June 2 to 12, Pan-Pacific Auditorium, Los Angeles. Designed to show commercial, residential, and industrial materials, products, techniques, and services, it is owned and sponsored by 13 industry associations and the Los Angeles Chamber of Commerce.

American Society of Refrigerating Engineers, 40 W. 40th St., New York City—Spring meeting, June 5 to 8, aboard ship during a cruise from Montreal on the St. Lawrence and Saguenay rivers.

Heating, Piping & Air Conditioning Contractors National Association, 1250 Avenue of the Americas, New York 20—60th annual meeting, June 8 to 10, Hotel Statler, Detroit.

The 4th International Gas Conference, June 13 to 17, London, England—Sponsored by the International Gas Union, comprised of national gas associations throughout the world. Additional information may be obtained from the American Gas Association, 420 Lexington Ave., New York 17.

American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York 10—1949 semi-annual meeting, June 20 to 22, Minneapolis.

American Society for Testing Materials, 1916 Race St., Philadelphia 3, Pa.—The fifty-second annual meeting will be held at the Chalfonte-Haddon Hall hotel, Atlantic City, June 27 to July 1. The first national meeting of the society on the west coast will be held at the Fairmont hotel, San Francisco, October 10 to 14.

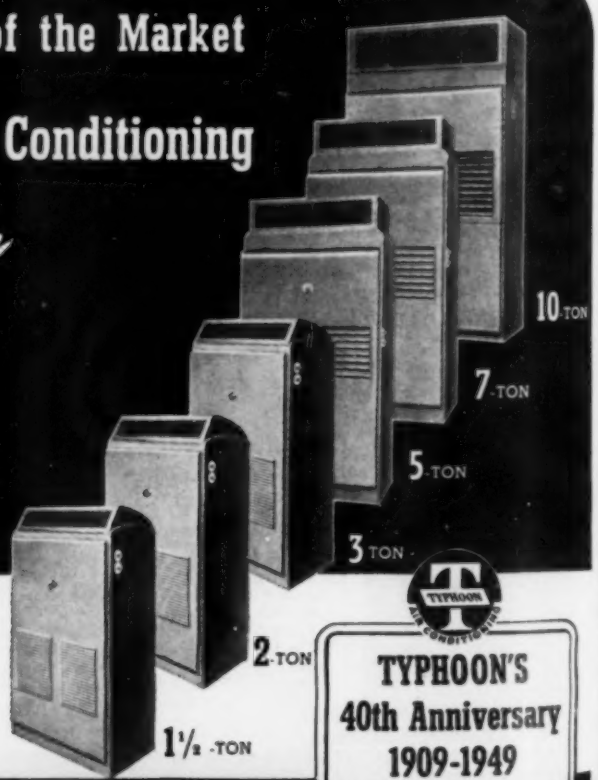
American Welding Society, 33 W. 39th St., New York 18—The 30th annual meeting and exposition will be held in Cleveland, the week of October 16.

Sixth All-Industry Refrigeration and Air Conditioning Exposition, November 14 to 18, Atlantic City—sponsored by the Refrigeration Equipment Manufacturers Association and managed by R. K. Hanson, show director, 1107 Clark Bldg., Pittsburgh 22. Over 75,000 sq ft of floor space will be used and 5200 hotel rooms have been reserved for this event.

Concentrate on the *Cream* of the Market

THE QUICK PROFIT RANGE in Air Conditioning

1½ Ton to *10 Ton*
PACKAGED UNITS



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Plus a line of: EVAPORATIVE CONDENSERS — 3-5-7 and 10-ton sizes. COMPRESSORS — ½-ton to 50-ton units. LOW SIDE UNITS — 3-5-7-10-Ton.

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You can do more work every day when you bend pipe with a

TAL Portable Bender

Work goes faster when you bend on the job — either original installation, radiant heat or repair. Fewer joints to leak in the future, less friction, more satisfaction for your customers and a bigger profit for you. Do as hundreds everywhere are already doing. Write for illustrated data bulletin. Dept. 13

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The Haney Electronic flame failure safety control panel, ready for installation, provides safe, sure and positive combustion control and will meet insurance requirements. It is so designed that the burner cannot operate unless a pilot flame of sufficient size to ignite the burner is present and the unit itself is functioning.

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Two-Piece Nozzle is easily cleaned without removal from piping

One-Piece Nozzle . . . for maximum uniform output at lowest pressure

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Stop Needless Waste

Pipe joints sealed with Key-Tite cannot leak. Key-Tite is a perfect seal on all lines carrying water, gas, low-pressure steam, compressed air, etc. It contains no lead or substitutes and does not affect the taste or odor of potable liquids. Joints are easily opened, too, for it will not freeze in the joint. Stop losing money through leaky pipe joints . . seal with Key-Tite.

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Here's positive, controlled ventilation at all times, regardless of wind or weather! Snow, ice, wind or rain have no effect upon the certified performance ratings of the POWER-FLOW. Exhausts more air per brake horsepower than other ventilators. May be used with or without duct system.

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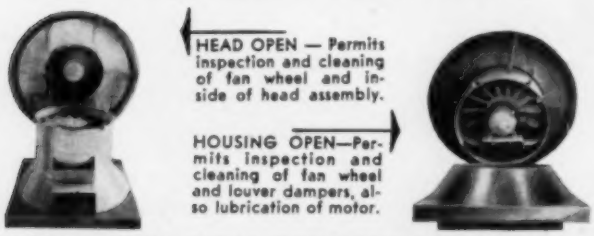
HEAD OPEN — Permits inspection and cleaning of fan wheel and inside of head assembly.
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Low streamlined design of the POWER-FLOW blends well with modern building design.

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Please send, without obligation, new POWER-FLOW Roof Ventilator Catalog No. DP-3-48.

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RECENT TRADE LITERATURE

For your convenience in obtaining copies of these bulletins, see coupon on page 207. If you write direct to the manufacturer, describe carefully what literature you want, as the number given first in each item is for use only when sending requests to Heating, Piping & Air Conditioning.

No. 7165: Adhesives for Use With Insulation—A 16 page manual on sealers, surface coatings, cements, mastics, emulsions, and other adhesives for use with various types of insulation in both industrial and marine applications is available from Benjamin Foster Co., 4607 W. Girard Ave., Philadelphia 31. Information is given on viscosity, coverage, flammability, heat flow, and resistance to acids and alkalis. Also included is a chart summarizing bonding and drying times, colors, temperature limits, types of thinner, and other information.

No. 7166: Adjustable Pipe Hangers and Supports—Catalog 10-D, released by Grinnell Co., Inc., 280 W. Exchange St., Providence 1, R. I., contains 90 pages on spring hangers, hanger assemblies, hanger rods, pipe supports, and miscellaneous items such as clips, expansion cases, extension pieces, sockets, couplings, hooks, turnbuckles, ceiling plates, etc. In addition to giving dimensions, prices, recommended loads, and other data on units designed for attachment to pipe and attachment to structures, example hanger load calculations are worked out for high temperature, high pressure piping systems. Also included are standard specifications for pipe hangers.

No. 7167: Air Sterilization, Freshening—Pittsburgh Chemical Laboratories, Inc., Pittsburgh, has released a four page folder containing a reprint of an article by Albert Q. Maisel on preventing the common cold through the use of triethylene glycol, published in Hygeia. Also described is "Pittairaide" a solution composed of this glycol and "Chlorofal." The former is used to kill airborne germs and the latter is used to dispel odors. A portable dispenser and a unit for use in conjunction with air conditioning and ventilating systems are described.

No. 7168: Centrifugal Pumps—Series 122 "Uni-Built" centrifugal pumps feature standard motors, a mechanical seal, a flexible coupling, and compact design. Offered by Bell & Gossett Co., Morton Grove, Ill., they are available as all-iron, bronze fitted, or as all-bronze units. Bulletin AZ-348 provides detailed information including dimensions, capacities, and recommendations for installation and operation.

No. 7169: Ceramic Tank Lining—Advantages of "Porceline" lining for commercial storage tanks are briefly covered in a four page folder issued by Apex Engineering Co., 75 E. Wacker Dr., Chicago 1. Said to expand and contract with the tank, it is recommended for use on iron, steel, or concrete. According to the company, it prevents corrosion, rust, pitting, scale, and similar problems. Another four page folder shows the application of the lining to a 500,000 gal water storage tank and also lists capacities in gallons and dimensions of cylindrical tanks.

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A minimum fuel reduction of 5%, elimination of nuisance smoke, fly ash and air polution is guaranteed on a money-back basis.

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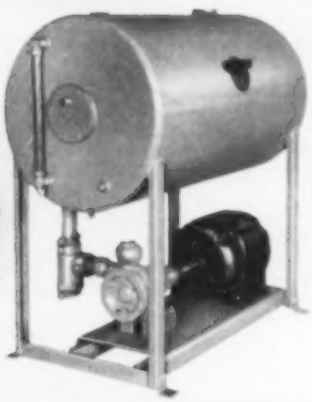

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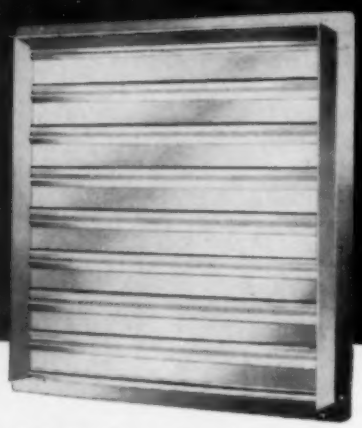

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Aluminum louvers open fully, permitting capacity fan operation. New heavy reinforcement strip adds strength and long life to the louvers, assures quiet operation and perfect counterbalance, prevents rattling. Deep shroud protects shutter from high winds. Tie-rod, brackets and bearings inside frame, not exposed to weather. Special finish resists corrosion. Many other features.

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Illustrations and details of the complete Air-Flo line.

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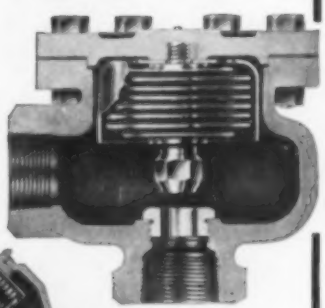
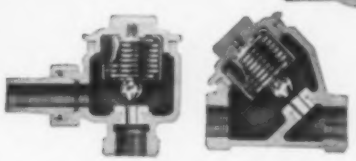
PRODUCTION INCREASES UP TO 30%

2 to 6 Times More Drainage Capacity Speeds Circulation





Bulletin 1047 or see Sweet's

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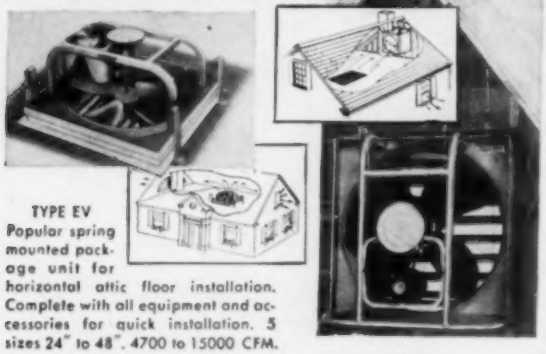

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No. 7170: Conditioning Crane Cabs—The Vol. 3, No. 3 issue of the house organ published by Dravo Corp., Dravo Bldg., Pittsburgh, features a resumé of a paper on air conditioning crane cabs by R. D. Darrah of the company. Presented at last year's convention of the Association of Iron and Steel Engineers, the paper gives data on temperatures and humidities encountered in various types of crane operation and also lists allowable concentrations of gases, vapors, mists, and dusts. A temperature chart indicates the great amount of radiant heat to which crane cab operators are subjected. Typical conditioning layouts and uses of the company's equipment are illustrated. Another article describes the use of the firm's direct fired air heaters in tempering air supplied to a foundry.

No. 7171: Convector-Radiators—Design and construction features, dimensions, and ratings of the type F convector-radiators offered by Fedders-Quigan Corp., Buffalo, N. Y., are contained in Catalog FC-1. Designed for use with steam and gravity or forced hot water heating systems, the heating elements are of copper tubes with mechanically bonded aluminum fins. Cabinets are for either free standing or semi-recessed installation.

No. 7172: Electronic Smoke Indicator for Overfire Jet Control—The photoelectric cell smoke indicator of Brooke Engineering Co., Inc., 4517 Wayne Ave., Philadelphia 44, has been adapted to control overfire air jets so as to prevent smoke. Features of the system, component parts, and applications are covered in Bulletin No. 20-A.

No. 7173: Fire Tube Boilers—Bulletin No. 125 has been issued by Federal Boiler Co., Inc., Granite & West Sts., Midland Park, N. J., and covers series FC and series FD steel, fire tube boilers designed for use with coal, gas, or oil. The FD units are offered in three sizes from 1500 to 2200 sq ft EDR steam.

No. 7174: Heating Control Service Manual—A 72 page illustrated handbook for the heating control installer and service man has been prepared by Minneapolis-Honeywell Regulator Co. It is said to apply to all oil, gas, and stoker fired systems and contains suggestions for quick and efficient servicing. Subjects covered include code identification, control circuits, thermostats, limit controls, trouble shooting, transformer specifications, etc. Free copies are available by writing to Walter Baak, manager of the company's wholesale division, 2753 Fourth Ave. S., Minneapolis. The booklet will also be available in quantities for 25 cents each.

No. 7175: Heat Transfer Coefficients for Reflective Insulation—A new handbook of insulation values is now available to engineers, builders, and architects upon written request to Silvercote Products, Inc., 161 E. Erie St., Chicago 11. Containing 108 pages, 12,852 coefficients, and 238 sketches, the book lists *U* values for various types of wall, floor and ceiling construction utilizing the company's reflective insulation. As is customary, coefficients are given in Btu per hr per sq ft per deg temperature difference. "Summer" values for ceilings are also given, and these data are based on 90 F air temperature below the ceiling and 130 F temperature above the ceiling. The coefficients, says the company, were calculated in accordance with the FHA procedure.

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EVERYONE who is boiler-wise will say they're "great" or "fine" when Brownell boilers are mentioned. This reputation has grown worldwide since 1855, the year our company was founded. Those pioneer Brownell boiler makers made Brownell a name which means honest, rugged, dependable equipment. We of today are carrying on the tradition.

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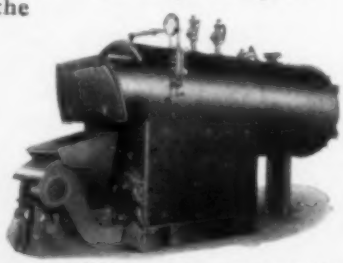
When you're in the market for boilers, stokers, and heat exchangers, be sure to get complete information on the Brownell line.



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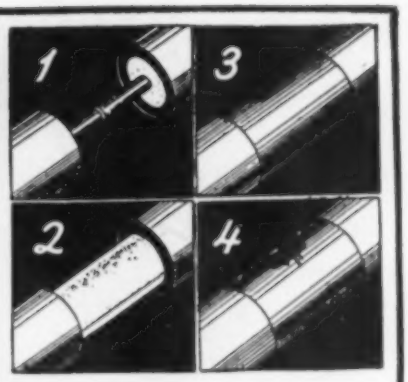

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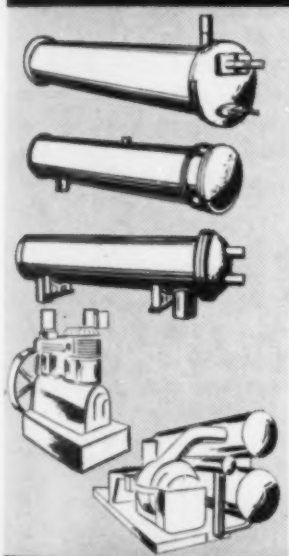

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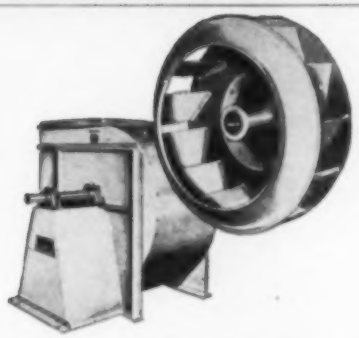

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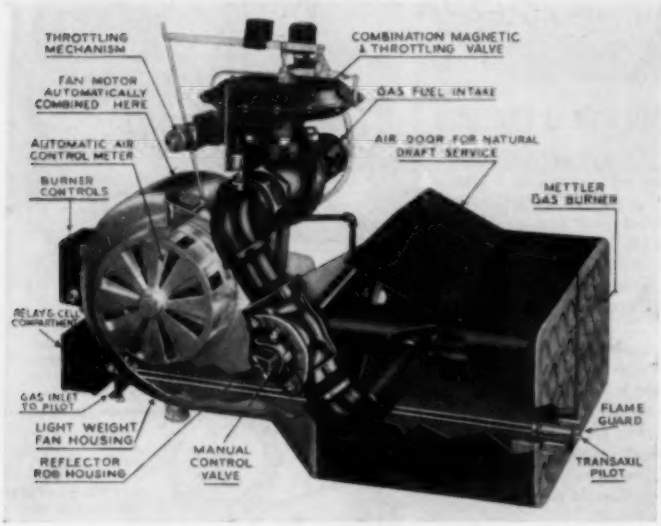

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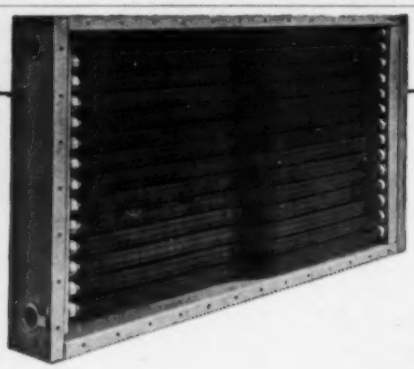

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No. 7176: Liquid Level Control—"Prob-O-trol" type X, described in Bulletin 9800, is a probe type liquid level control either for differential or specific level operation. It is available from Automatic Control Co., 1005 University Ave., St. Paul 4, Minn., for many applications in process industries, etc. and is particularly recommended where corrosive conditions or sanitary regulations prevail.

No. 7177: Mechanical Seals for Chemical Pumps—The line of seals offered by Worthington Pump and Machinery Corp., Harrison, N. J., is now available for the company's type GG "Worthtite" pumps, up to 1200 gpm capacity. The seals and pumps are recommended for many processes involving chemicals. Bulletin W-350-B10 describes these type EA seals.

No. 7178: Modulating Controls—The regulators which derive their motive power from the expansion of a thermo-sensitive liquid confined in the thermal system and which are available from Fulton Sylphon Div., Robertshaw-Fulton Controls Co., Knoxville 4, Tenn., are described in a new bulletin. Typical applications of control valves, temperature regulators, safety regulators, room thermostats, and radiator valves in heating, ventilating, and air conditioning systems are featured.

No. 7179: Portable Soldering Tool—A new "Thermo-Grip" soldering tool, operating on the resistance heating principle and designed for use on small tubing in refrigeration work, small motors, switchboards, and other electrical equipment, is described in Catalog Section C-1, issued by Ideal Industries, Inc., Sycamore, Ill. Touching the work with the portable tool completes the secondary power circuit and causes the part touched to heat quickly.

No. 7180: Pressure Drop Chart for Steam Separators—In order to facilitate the selection of its steam separators for various conditions, Wright-Austin Co., 315 Woodbridge St., Detroit 26, has prepared a pressure drop chart. Also included as a part of the chart is a method for determining the pressure drop through an oil separator for an exhaust steam line and three typical examples are worked out. Charts may be obtained from the company by engineers now engaged in using steam generating equipment.

No. 7181: Pressure Reducing and Regulating Valves—Single seated, diaphragm actuated reducing valves for either steam or water service, a pilot operated auxiliary steam pressure regulator, and double seated reducing valves for steam service are described and illustrated in Catalog S-47, issued by Watts Regulator Co., Lawrence, Mass. Capacities are tabulated in a flow chart.

No. 7182: Radiant Heating Controls—Sarcotherm Controls, Inc., Empire State Bldg., New York 1, has issued Technical Bulletin No. 1, the first in a series on controls for radiant heating systems. Four basic control systems are explained and illustrated and new thermostats recently developed by the company are described.

No. 7183: Refrigeration, Air Conditioning Instruction—Utilities Engineering Institute, 2525 N. Sheffield Ave., Chicago 14, since 1927 a private vocational trade and

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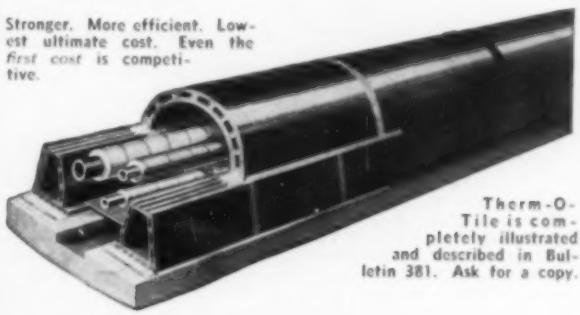

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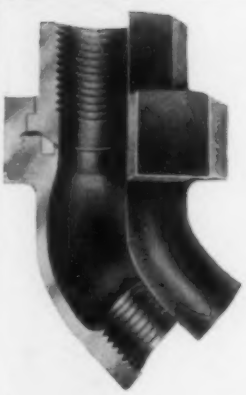

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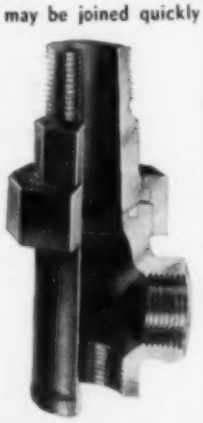

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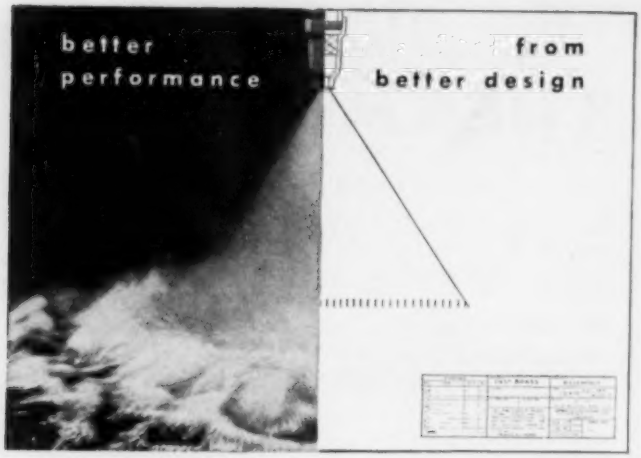

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technical school, has issued a new 72 page catalog giving detailed information on courses offered, the facilities, and personnel of the school. Several new courses in air conditioning and refrigeration are described.

No. 7184: Refrigeration Hot Gas Defrosting System—Colored diagrams are used in Bulletin DF-100 to explain the operation of the automatic system offered by Patterson-Kelley Co., Inc., 101 Park Ave., New York 17. Requiring but four pipe connections when added to an existing refrigeration system, it is based on the company's "Slug Eliminator," a combination liquid trap and heat exchanger used to evaporate liquid in the suction line and to sub-cool the liquid refrigerant.

No. 7185: Roof Sprayers—A chart designed to show the possible reductions in cooling loads by spraying roofs with water has been prepared by April Showers Co., Inc., 4126 Eighth St., N. W., Washington 11, D. C.

No. 7186: Self Priming Centrifugal Pumps for Petroleum Products—Engine driven models in sizes from 1½ to 6 in. and with capacities from 15 to 2000 gpm, together with electric driven models in the same sizes are described and illustrated in a 12 page bulletin (No. PM48) released by Marlow Pumps, Ridgewood, N. J. A number of selection tables are included and data are tabulated on specific gravity, viscosity, head equivalents of various liquids, maximum practical suction lifts, friction loss in feet of liquid per 100 ft of clean steel pipe, pressure losses per 100 ft of rubber hose, and friction losses of fittings and other equipment.

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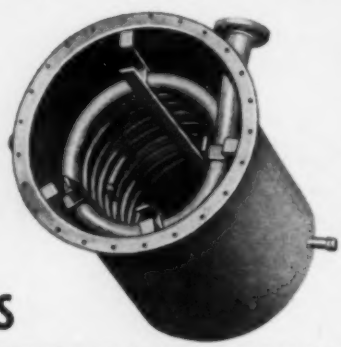

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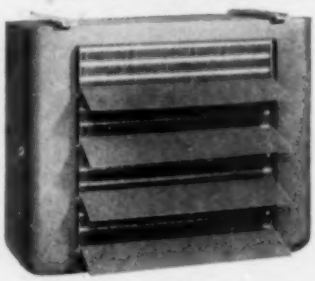

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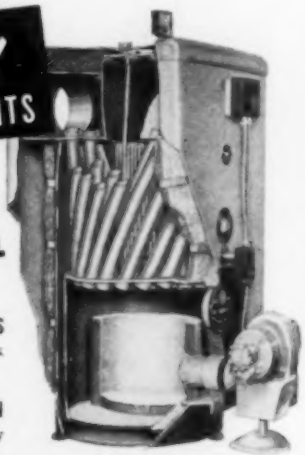

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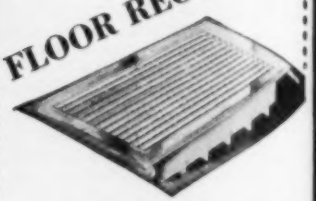

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